United States Court of Appeals

for the Second Circuit

EXHIBITS

Cal #
715

75-7362

United States Court of Appeals
FOR THE SECOND CIRCUIT

Docket No. 75-7362

PERMA RESEARCH & DEVELOPMENT COMPANY,

Plaintiff-Appellee,

—v.—

THE SINGER COMPANY,

Defendant-Appellant.

ON APPEAL FROM THE UNITED STATES DISTRICT COURT
FOR THE SOUTHERN DISTRICT OF NEW YORK

UNITED STATES COURT OF APPEALS
FILED
DEC 29 1975
A. DANIEL FUSARO, CLERK
SECOND CIRCUIT

EXHIBIT BINDER

VOLUME I (Pages E-1 through E-453)
DESIGNATED PLAINTIFF'S EXHIBITS (5A-57)

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9	Patent #3,325,226 Braking System and Anti-Skid Means Therefor, dated June 13, 1967	E-18
10	Patent #3,330,113 Power Brakes, dated July 11, 1967	E-36
11	Patent #3,383,482 Switch Devices, dated May 14, 1968	E-53
12	Patent #3,459,204 Anti-Skid Device Valves, dated Aug. 5, 1969	E-69
13	Patent #3,477,765 Acceleration Responsive Devices for Anti-Skid Units, dated Nov. 11, 1969	E-85
14	Letter to Frank Perrino from Keith Heffer, Sales Manager, The Singer Company of Canada, Ltd., dated Jan. 21, 1964	E-101
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18A	Complete Brochure Described in Plaintiff's Exhibit 18	E-107
19	Motor Vehicle Research Report No. 13, Published at Lee, New Hampshire, 1964	E-135
20	Ads, Brochures, Article Reprints of the Perma Anti-Skid Device	E-152
21	Cadillac Engineering -- 1963-1964, Perma Anti-Skid Tests	E-164
22	Report of C. R. Fricke of Singer on Meeting Held March 13, 1964 Concerning Perma's Anti-Skid Control	E-165
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29	Agreement Between Perma Research and Development Company and Clarelen Importers, Ltd., dated June 10, 1963	E-340
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31	Agreement Between Perma Research and Development Company and Richard H. Travers and John W. Travers of Quincy, Mass., dated Feb. 12, 1964	E-360
32	Agreement Between Perma Research and Development Company and Angelo Di Spirito, Jr. and Christie Vitolo	E-366
34	Agreement Between The Singer Company of Canada,Ltd. and O-Matic Auto Products, Ltd., dated Jan. 1, 1965	E-372
35	Contract Between Perma Research and Development Company and The Singer Company, dated June 18, 1964	E-378
38A	Group Memoranda Entitled: (a) Meeting Between Perma and Singer Personnel, dated Sept. 2, 1964 (b) Meeting Between Perma, American Optical and Singer Personnel, dated Sept. 3, 1964 (c) Meeting Between Perma, Singer and American Optical Personnel, dated Sept. 8, 1964 With Results of Test (Sept. 8, 1964)	E-403
39	Letter from A. Romel to J. Fecko: "Status Report--Perma Anti-Skid Device", dated Sept. 17, 1964	E-417
47	Minutes of Meeting of Sept. 24, 1964--Subject: Perma Anti-Skid Unit	E-420

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49	Memorandum to C. W. Broeker from A. Romel--Subject: Status Report "Perma", dated Sept. 28, 1964	E-425
53	Memorandum to J. Fecko from A. Romel Re: "Perma Sensing Units", dated Dec. 4, 1964	E-431
54	Memorandum Re: Discussion Between Perma & Singer Personnel, dated July 17, 1964	E-432
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89	Memorandum to B. C. Person from David McKearnan, dated Aug. 16, 1965	E-553
90A	Memorandum to D. P. Kircher from D. D. Marsden Enclosing Perma Anti-Skid Status Report and Recommendations, dated Aug. 30, 1965	E-557

Plaintiff's Exhibit No.	Description of Document	Exhibit Binder Page
92	Letter from B. C. Person to Frank Perrino, dated Sept. 9, 1965	E-568
94 (pages 3-5)	Portion of Report Letter to B. C. Person from C. W. Broeker with Report Attached, dated Sept. 1, 1965	E-570
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99	Memorandum to A. di Scipio from B. C. Person - "Recommended Course of Action for Perma Anti-Skid Control Program", dated Nov. 9, 1965	E-576
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120	Memorandum to D. D. Marsden from R. S. Torello, dated Dec. 21, 1964	E-603
121	SAE Booklet "Evolution of Sure-Track Brake System"	E-609
122	Portion of Plaintiff's Pre-Trial Memorandum, Perma Research & Development Co. v. The Singer Co. and Additional Reasons Rendering the Perma Anti-Skid Device Unmarketable, dated Oct. 6, 1972	E-628
123	Simulation of Perma Anti-Skid Device (24:1)	E-636
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125	Simulation of Perma Anti-Skid Device (18:1)	E-638
127 (first 3 stops)	Computer Simulation of Perma Anti-Skid Device, Gear Ratio 24:1	E-639
128 (first 3 stops)	Computer Simulation of Perma Anti-Skid Device, Gear Ratio 12:1	E-645

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129 (first 3 stops)	Computer Simulation of Perma Anti-Skid Device, Gear Ratio 18:1	E-651
130	Anti-Skid Control: General Description of Major Assemblies, dated May 7, 1965	E-657
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132	Vehicle Equations of Motions by A. DeVilliers	E-689
133	Numerical Constants Used In Vehicle Simulation	E-691
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135	Numerical Constants in the Simulation of the Perma Anti-Skid Device	E-697
136	SAE Paper Entitled "Tire-Road Friction Measuring System - A Second Generation" by G. Goodenow, T. Kolhoff, F. Smithson	E-699
140	Statistical Abstract of the United States: Motor Vehicles 1972	E-708
142	Diagram of Cam Gear	E-713
143	Road Surfaces Denoted by 60 MPH Sliding Coefficient of Friction	E-714
144	Annual Report, 1965, The Singer Company	E-718
147	The Singer Company: Experimental and Test Data Sheet, dated Nov. 30, 1964	E-745
150	Application for Patent #322,716 Entitled "Braking System and Anti-Skid Means Therefor", filed Nov. 12, 1963	E-746
154 (at 23)	Ford Sure-Track Manual	E-819
155 (Chapter 5)	Chrysler Service Manual, 1972	E-820

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156 (Chapter 5)	Cadillac Service Manual, 1973	E-905
160	Plot of A. DeVilliers' Computer Simulation at 60 MPH, .75 mu, 12:1 gear ratio	E-982

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163	1969 New Model Training: Brake Systems Workbook	E-990

II. Defendant's Exhibits Received In Evidence

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1	Soundtrack of Perma Promotional Film (Transcribed Nov. 14, 1973)	E-1017
8	Three-part Cardboard Display Entitled "Perma Anti-Skid Control"	E-1027
10	History of the Perma Research and Development Company, 1952-1962	E-1030
11	Letter from G. J. Salustro of Ford Motor Company to G. Lee Dienes of Perma Research and Development Company, dated June 15, 1964	E-1039A
12	1964 Ford Test Results	E-1040
20	Letter from L. H. Nagler of American Motors Corporation to Roger Levesque, dated June 26, 1964	E-1042
25	Special Report No. 2, Confidential	E-1044
36	Two Field Reports: 1) From Roger Levesque and Tom Perrino to F. A. Perrino, R. D. Mainelli and J. Krawczyk. Installation of Steel Drive Gears in Connecticut State Police Cruisers. Westport Barracks and Bethany Barracks. 2) From Tom Perrino to F. Perrino, R. Mainelli, J. Krawczyk and L. Dienes. Connecticut State Police - Westport Barracks, c/o Safety Service Corp. Replacement of Plastic Speedometer Gear with Steel Replacement Gear, dated July, 1964	E-1046
44	Report from Roger Levesque to F. A. Perrino, R. D. Mainelli, J. Krawczyk, L. Dienes and S. Karp, dated Aug. 27, 1964	E-1050
53	Affidavit of Frank A. Perrino, dated Nov. 18, 1969	E-1051
102	Letter from J. Gomes to R. A. Kloby, dated July 23, 1964	E-1067
111	Memorandum from A. Romel to C. W. Broeker re: "Gear Boxes" for Perma Anti-Skid Units, dated Oct. 26, 1964	E-1068
114	Memorandum from A. Romel to C. W. Broeker re: Summary of Gear Box Test Results, dated Nov. 16, 1964	E-1073

Defendant's Exhibit No.	Description of Document	Exhibit Binder Page
115	Memorandum from A. Romel to C. W. Broeker re: Summary of Gear Box Test Results, dated Nov. 24, 1964	E-1076
118	Memorandum from A. Romel to C. W. Broeker re: Summary of Gear Box Life Test Results, dated Dec. 3, 1964	E-1080
191	Letter from Cecil S. Morris to S. J. Murphy re: Attached Report Outlining Singer's Position Under Manufacturing Licensing Agreement with Perma Research and Development Company, dated Oct. 21, 1964	E-1083
195 (pages 1-4)	Letter from S. Karp to Robert Patten of The Singer Company re: Estimated Budget Projection, dated Nov. 30, 1964	E-1093
197	Letter from S. Karp to Ed Wright of Car Controls, Inc. Instructing Him to Withhold Perma Anti-Skid from the Market, dated Dec. 11, 1964	E-1098
233	Memorandum from A. Romel to C. W. Broeker re: Summary of Gear Box Life Test Results, dated Dec. 23, 1964	E-1099
234	Memorandum from A. Romel to C. W. Broeker re: Endurance Testing of Sensing Unit, dated Dec. 28, 1964	E-1102
236	Memorandum from A. Romel to C. W. Broeker re: Summary of Endurance Tests on the Sensing Unit, dated Feb. 1, 1965	E-1103
242	Memorandum from A. Romel to C. W. Broeker re: Summary of Endurance Tests on the Sensing Unit, dated Feb. 10, 1965	E-1105
313	Memorandum from H. Hnojowy to R. A. Kloby re: Testing of the Pressed Metal Filters, dated July 12, 1965	E-1107
323	Report from P. Urban to A. Romel Entitled, "Effect of Low Temperature on .030/.032" Diameter Metering Valve," dated Dec. 1, 1965	E-1109
336	Letter from A. Romel to B. C. Person re: The Ford Motor Company Trip, dated March 15, 1966	E-1153
380	Letter from S. Karp to R. R. Patten re: Need to Replace Units Installed on Consumers', Stockholders' and Dealers' Cars, dated Dec. 28, 1964	E-1155
409	Document Entitled "Perma Research and Development Co."	E-1156

Defendant's Exhibit No.	Description of Document	Exhibit Binder Page
417	Memorandum from R. A. Kloby to C. W. Broeker re: Letter Dated Jan. 7, 1965 from P. Phelps to S. J. Murphy	E-1166
479	Organization Announcement from D. P. Kircher, President of Singer, Announcing the Establishment of a New Operating Division to Bring Together Most of the U.S. Units Primarily Concerned with Consumer Products, dated June 25, 1965	E-1170
487	Memorandum from B. C. Person to A. di Scipio re: Attached Study Plan for the Anti-Skid Control Prepared for the Use of the Study Team, dated July 26, 1965	E-1173
502	Letter from R. A. Wolf, Head, Transportation Research Dept. of Cornell Aeronautical Laboratory, Inc., to B. C. Person re: Suggestion for a Three-phase Program of Research, dated Sept. 1, 1965	E-1178
529	Letter from R. A. Kloby to B. C. Person re: Attached Engineering Evaluation of the Latest Modifications Submitted by Perma to Provide a "Fail-Safe" Feature in the Anti-Skid Control Unit, dated Nov. 16, 1965	E-1180
529A	"Fail-Safe Analysis of Piston and Check Valve Modification", dated Nov. 10, 1965	E-1195
530	Memorandum from the Perma Anti-Skid Control Task Force to A. di Scipio re: Recommended Course of Action for the Perma Anti-Skid Control Program, dated Nov. 18, 1965	E-1196
577 (excluding appendices)	Booklet Entitled "Technical Status Report - Perma Anti-Skid Device", dated July 22, 1965	E-1202
595	Investment and Expenditures for Perma Anti-Skid Device	E-1227
603	Letter from R. A. Kloby to B. C. Person re: Perma Program Inventory Attachments, dated Oct. 20, 1965	E-1228
618	Cancelled Singer Check in the Amount of $85,000.00 to the Order of Monitor Enterprises, Inc., dated Sept. 1, 1965	E-1230
653 (Cover, p.7 "Advanced Control Concepts" thru p.11 to "Adaptive Steering")	SAE Document #690260, "Technological Development Directing the Path to Fully Automatic Vehicle Control Systems", by S. I. MacDuff, J. G. Rivard, E. J. McGlinn	E-1231
655	SAE Document #710248, "The Chrysler 'Sure-Brake' - The First Four-Wheel Anti-Skid System", J. W. Douglas, T. C. Schafer	E-1237

Defendant's Exhibit No.	Description of Document	Exhibit Binder Page
656	Executed Copy of Technical Services Contract Between Perma Research and Development Company and The Singer Company, dated Dec. 21, 1964, With Cover Letter from F. A. Perrino, dated Dec. 21, 1964	E-1248
670	Due to Its Excessive Size, by Stipulation Dated Aug. 7, 1975, DX 670 (Blueprint of Vacuum Container DC 17-342066-900X) is in the Possession of Counsel for Defendant. DX 670 Will be Produced for the Court's Inspection Upon Request.	E-1253
686	"Perma Anti-Skid Detail of Start-Up Manufacturing Expenses July 1 - Dec. 31, 1964"	E-1254
687	Employee History and Compensation Records (Top Page: Albert E. Romel)	E-1256
688 (p. 909 thru p. 910 to "Measurement Methods")	Article on Measurement of Tire Brake Force Characteristics as Related to Wheel Slip (Antilock) Control System Design by J. L. Harned, L. E. Johnston and G. Scharpf	E-1270
689 (p. 298 "Abstract", p. 299 "Braking and Safety" thru p. 306, p. 317 "Automatic Regulation Devices" thru p. 318)	Article on Automatic Brake Proportioning Devices by Guiseppe Puleo	E-1272
700 (at 1, 123-124)	1973 Edition of Ford Brochure Entitled "Car Buying Made Easier"	E-1283
708 (cover, pp. 1, 13, 15, 22, 27, 31, 34, 35, 40, 44)	Modified Plaintiff's Exhibit 129	E-1287

III. Defendant's Exhibits Offered In Evidence At Trial And Rejected

13 (id)	Engineering Appraisal of "Perma" Anti-Skid Device, dated April 9, 1964	E-1298
335 (id)	Memorandum from B. C. Person to S. Boriss, C. W. Broeker and R. H. Elman re: Discussions of Messrs. Romel and Person with Kelsey-Hayes, with Attachments, dated Feb. 15, 1966	E-1302
337 (id)	Engineering Reports from The Ford Motor Company to A. E. Samson of Ford re: The Effectiveness of the Anti-Skid Control Unit as Evaluated by Ford	E-1311

Defendant's Exhibit No.	Description of Document	Exhibit Binder Page
510 (id)	Memorandum from J. A. Lundin to J. W. Momberg of Singer re: Meeting at Denville Research Laboratory on Sept. 30, 1965	E-1326
559 (id)	Memorandum from A. Romel to B. C. Person re: Trip Report to Ford Motor Company, dated Feb. 17, 1966	E-1331
560 (id)	Letter from B. C. Person to C. F. Bruder re: Information Reviewed with D. P. Kircher of Singer, dated Feb. 25, 1966	E-1336
658 (id)	Letter from S. Linderman to P. DeRensis re: Plaintiff Counsel's Request for Perma Device - First Unit Delivered, dated May 16, 1973	E-1339
659 (id)	Letter from S. Linderman to P. DeRensis re: Plaintiff Counsel's Request for Perma Device - Second Unit Delivered, dated May 22, 1973	E-1340
684 (id)	Letter from H. J. Policinski, Patent Attorney for Kelsey-Hayes Co., to J. K. Lagemann re: Every's Testimony, dated June 6, 1974	E-1341

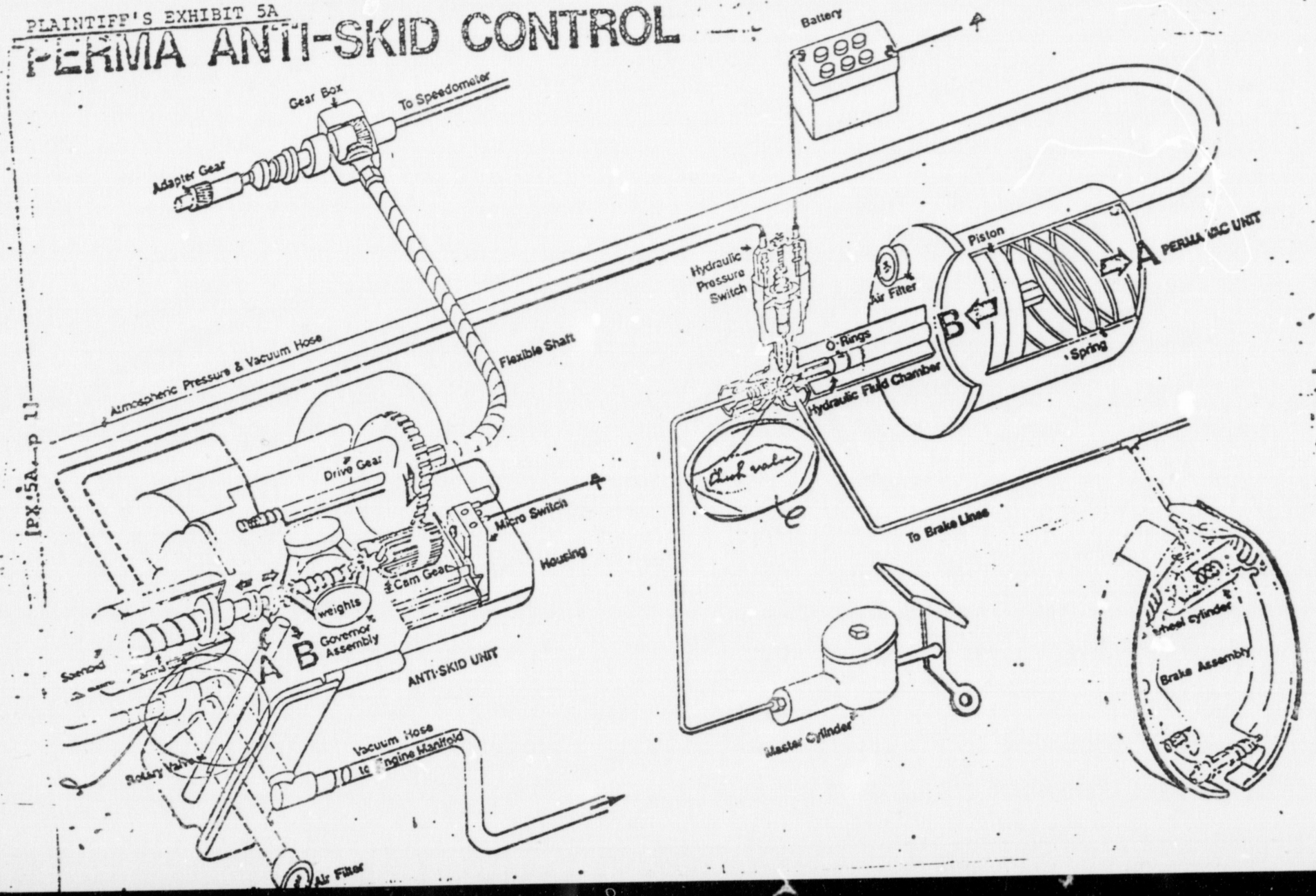
IV. Court's Exhibits

Exhibit No.		
1	Letter of P. R. Grand to J. K. Lagemann, dated Nov. 30, 1973	E-1343
5, p. 6 (p. 1 of Ex. A to Ct.'s Exhibit 5)	Portion of Partially Executed Stipulation of Facts	E-1345
6	Fully Executed Stipulation of Facts, dated June 24, 1974	E-1347

PLAINTIFF'S EXHIBITS RECEIVED IN EVIDENCE AT TRIAL

PLAINTIFF'S EXHIBIT 5A

PERMA ANTI-SKID CONTROL



[PX 5A-p 1]

[PX 5A, p 1]

E-1

NO MATERIAL ON PAGE E-2

PLAINTIFF'S EXHIBIT 8

April 14, 1959 F. A. PERRINO 2,881,879

ACCELERATOR BRAKE CONTROL

Filed March 15, 1956 5 Sheets-Sheet 1

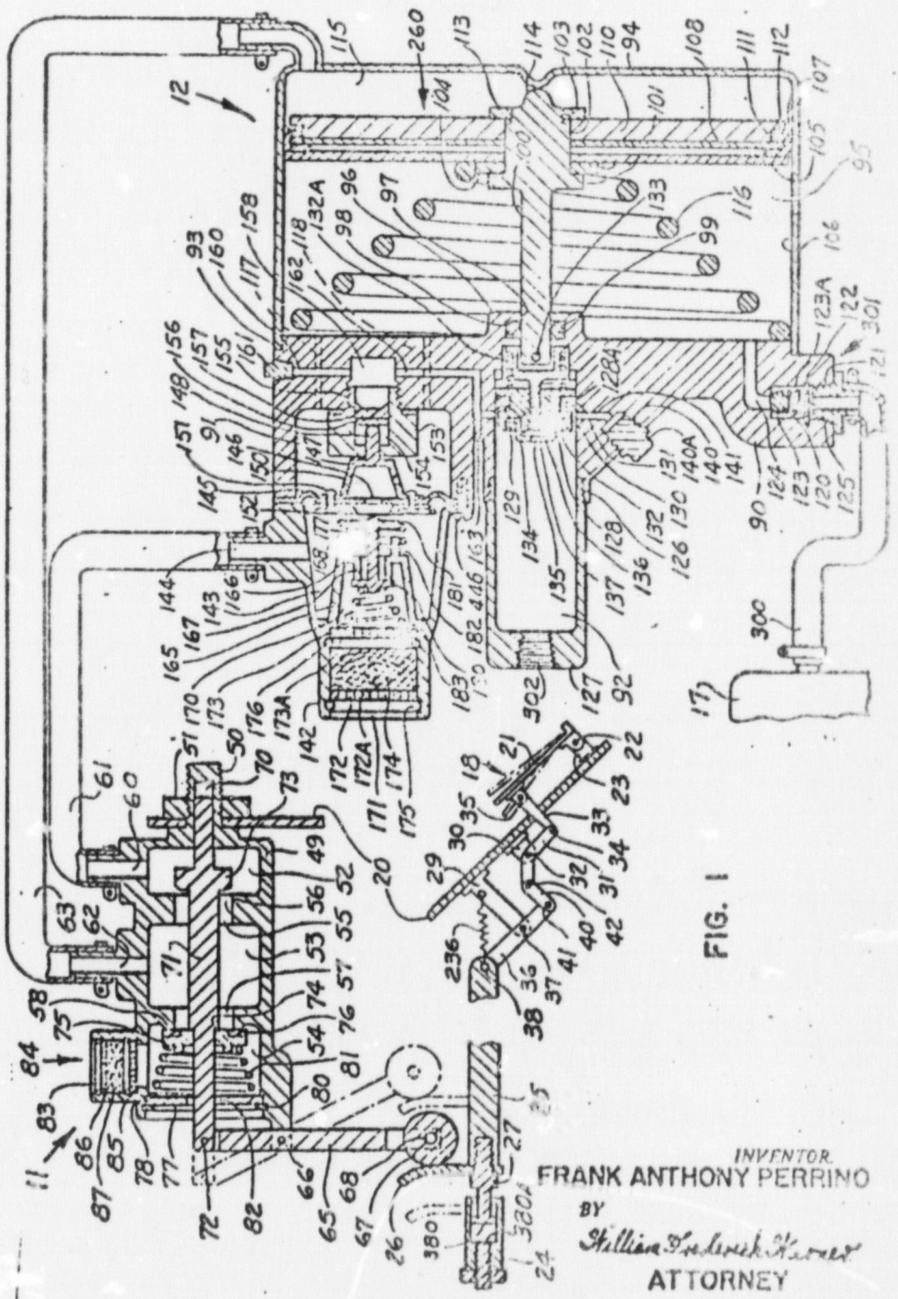


FIG. 1

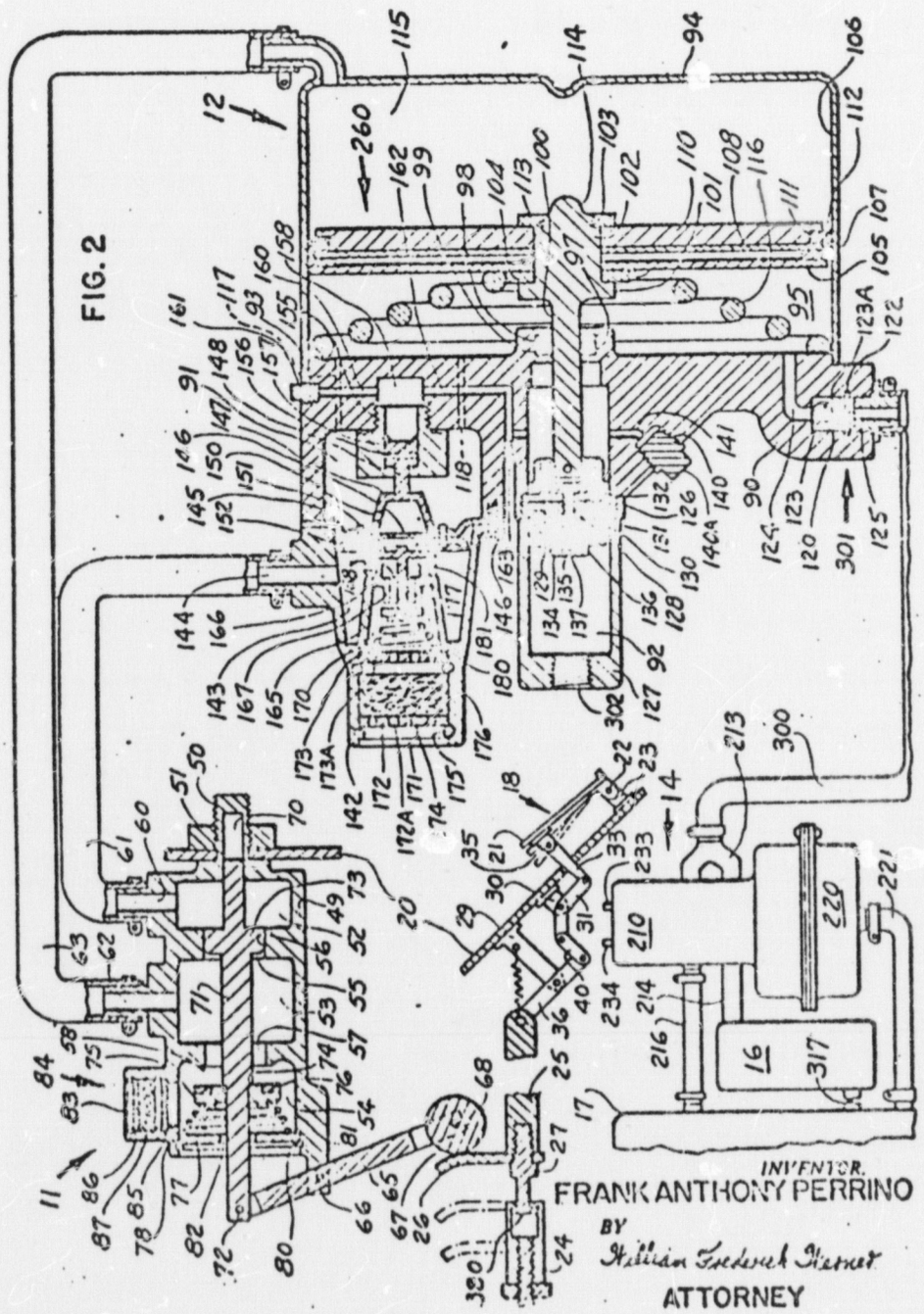
INVENTOR.
FRANK ANTHONY PERRINO
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William Frederick Kiener
ATTORNEY

April 14, 1959 F. A. PERRINO 2,881,879

ACCELERATOR BRAKE CONTROL

Filed March 15, 1956 5 Sheets-Sheet 2

FIG. 2



INVENTOR.
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[PX 8, p 3]

April 14, 1959 F. A. PERRINO 2,881,879

ACCELERATOR BRAKE CONTROL

Filed March 15, 1956 5 Sheets-Sheet 3

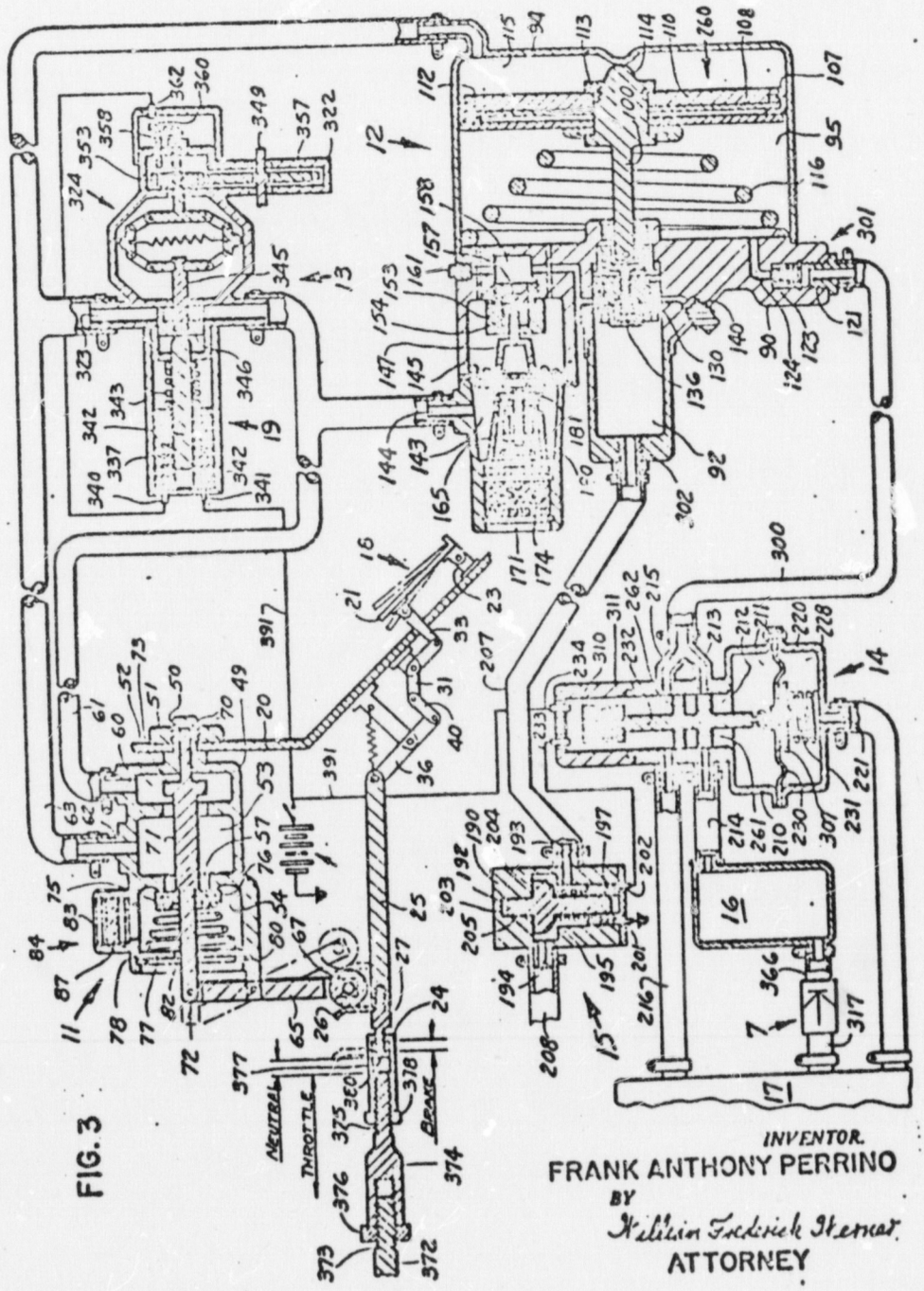


FIG. 3

INVENTOR.
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William Frederick Stewart
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[PX 8, p 3]

April 14, 1959 F. A. PERRINO 2,881,879

ACCELERATOR BRAKE CONTROL

Filed March 15, 1956 5 Sheets-Sheet 4

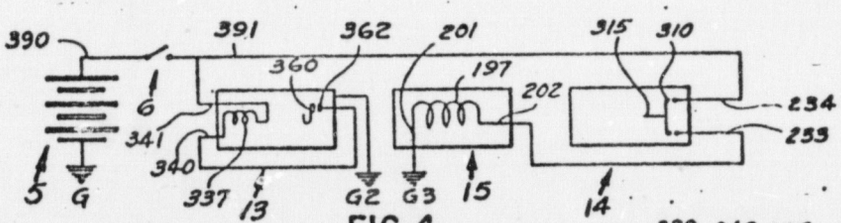


FIG. 4

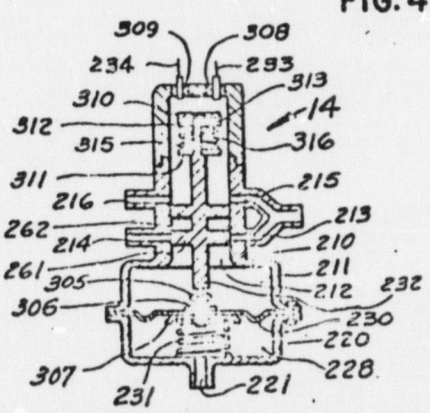


FIG. 5

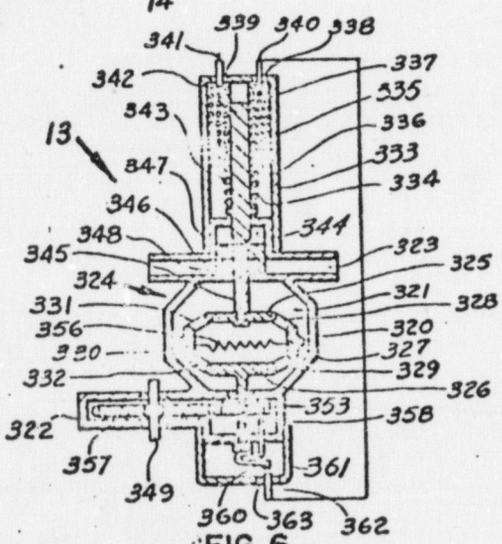


FIG. 6

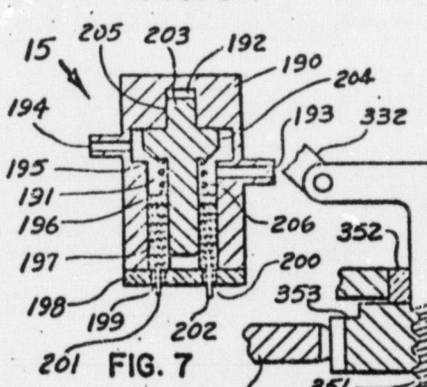


FIG. 7

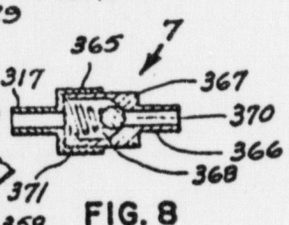


FIG. 8

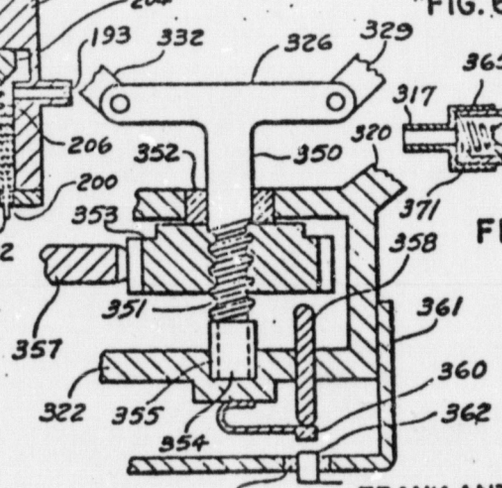


FIG. 9

INVENTOR.
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BY
William Frederick Varner
ATTORNEY

[PX 8, p 5]

April 14, 1959 F. A. PERRINO 2,881,879

ACCELERATOR BRAKE CONTROL

Filed March 15, 1956 5 Sheets-Sheet 5

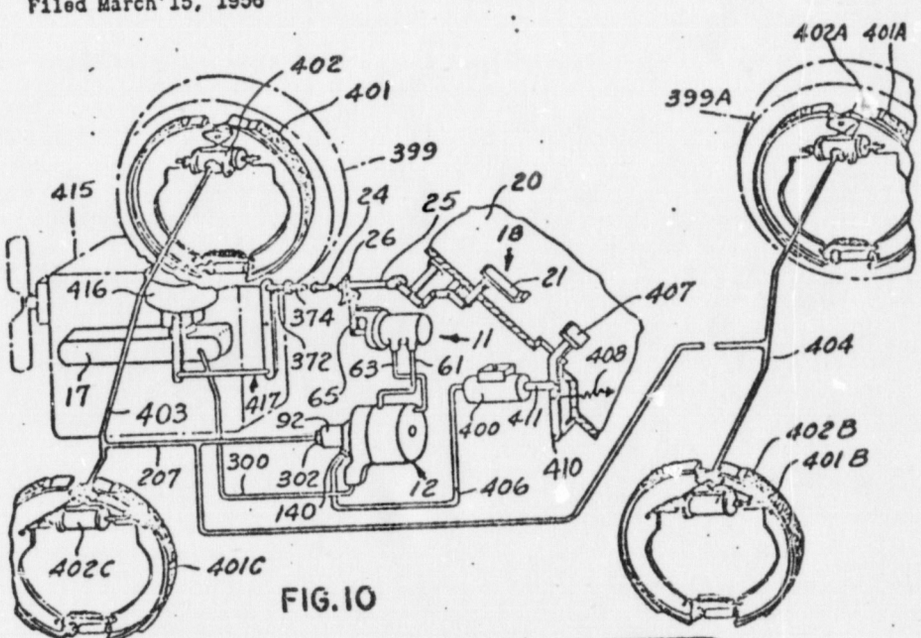


FIG. 10

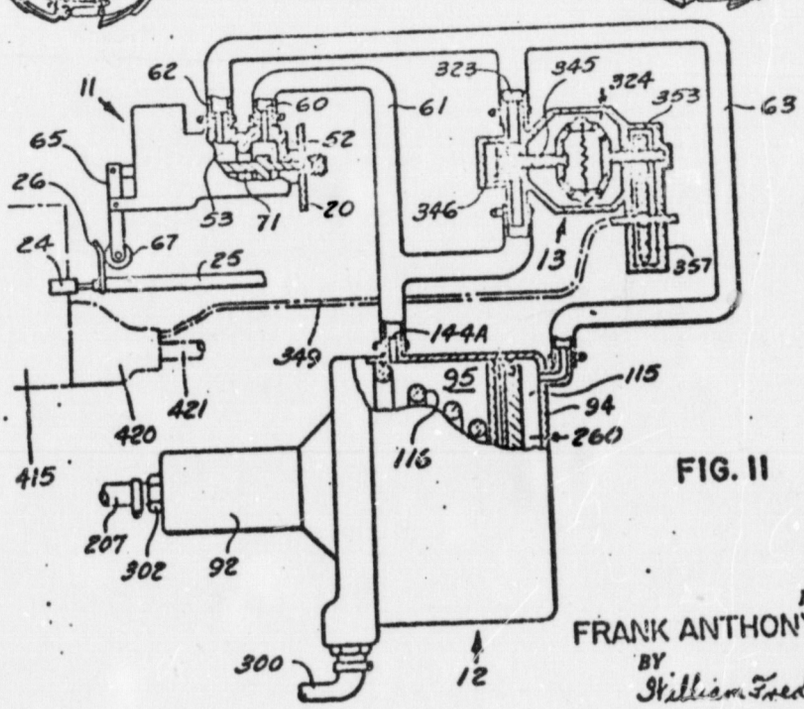


FIG. 11

INVENTOR.
FRANK ANTHONY PERRINO
BY
William Frederick Hennet
ATTORNEY

United States Patent Office

2,881,879

Patented Apr. 14, 1959

2,881,879

ACCELERATOR BRAKE CONTROL

Frank Anthony Perrino, North Providence, R.I.

Application March 15, 1956, Serial No. 571,734

22 Claims. (Cl. 192—3)

This invention relates to a power braking mechanism for motor vehicles and more particularly to a power braking mechanism under control of the accelerator pedal.

One of the objects of the present invention is to provide a power braking mechanism for moter vehicles which is under control of the accelerator pedal.

Another object of the present invention is to eliminate the conventional braking pedal on a motor vehicle and have the braking pedal function under control of the accelerator pedal.

Another object of the present invention is to provide an accelerator brake control which reduces the time interval between sensing the necessity for applying brakes and actually applying the brakes on a motor vehicle.

And still another object of the present invention is to provide an accelerator brake control with mechanism operating as an anti-skid device.

And still another object of the present invention is to provide an accelerater brake control with an emergency source of vacuum power.

And still another object of the present invention is to provide an accelerater brake control which may be operated in conjunction with the normal power brake pedal.

With these and other objects in view, the invention consists of certain novel features of construction which will be more fully described and particularly pointed out in the following specification and claims.

Like reference numerals refer to like parts in the accompanying drawings in which:

Figure 1 is a transverse sectional view of the accelerator brake control showing the power braking mechanism in neutral position.

Figure 2 is a view similar to Figure 1 showing the power braking mechanism in actuated position as when the brakes are applied; with a vacuum supply tank and an emergency valve added to the mechanism of Figure 1.

Figure 3 is a transverse cross-sectional view of a modified form of accelerator brake control showing safety features added to the mechanisms shown in Figures 1 and 2.

Figure 4 is a wiring diagram pertinent to the mechanism shown in Figure 3.

Figure 5 is a transverse cross-sectional view of the emergency valve.

Figure 6 is a transverse cross-sectional view of the anti-skid device.

Figure 7 is a transverse cross-sectional view of the solenoid check valve.

Figure 8 is a transverse cross-sectional view of the check valve.

Figure 9 is an enlarged fragmentary transverse cross-sectional view of one end of the anti-skid device shown in Figure 6.

Figure 10 is a schematic perspective view of a motor vehicle showing the braking system connected with the new and improved accelerator brake control.

Figure 11 is a side elevational view, with sections broken away for clarity, showing a modified form of anti-skid device and a modified form of power unit.

The present invention is adaptable to the conventional motor vehicle having hydraulic brakes and a fluid drive transmission.

The present invention as illustrated in Figure 1 shows a control valve 11, a power unit 12, an engine manifold 17 and an accelerator or gas pedal 18.

The control valve 11 and power unit 12 are shown with the movable parts in position when the motor vehicle is underway and the power brakes are inoperative.

Figure 2 illustrates the moving parts of control valve 11 and power unit 12 in position when the power brakes are in applied or operative position. In addition Figure 2 shows the added feature of an emergency valve 14 and a vacuum supply tank 16 added to the new and improved power braking mechanism illustrated in Figure 1.

Figure 3 illustrates additional safety operating features added to the new and improved power braking mechanism illustrated in Figures 1 and 2 consisting of an anti-skid unit 13, a solenoid check valve 15 and a solenoid 19 operatively connected to the anti-skid unit 13.

The floor board 20 of a conventional motor car is shown in Figures 1, 2 and 3. Accelerator pedal 21 is pivotally mounted at 22 to a bracket 23 which is fixed to floor board 20. The control sleeve leading to the carburetor is shown at 24. An intermediate link 25 provided with a flange 26 is connected to control sleeve 24 by means of stud 27.

A bracket 30 is fixed to floor board 20. A lever 31 is pivotally mounted at 32 to bracket 30. A link 33 is pivotally connected at 34 to lever 31 and is also pivotally connected at 35 to accelerator pedal 21. A toggle linkage consists of a lever 36 pivotally mounted at 37 to bracket 29 fixed to floor board 20 and is pivotally connected at 38 to intermediate link 25. A lever 40 is pivotally mounted at 41 to lever 36 and is pivotally mounted at 42 to lever 31.

It will be noted that when a driver's foot depresses accelerator pedal 21 it will pivot lever 31 through link 33 to pivot lever 36 through lever 40 to push intermediate link 25 forward so that control sleeve 24 will allow more gas to be fed from the carburetor and thereby cause the motor car to move forward from a neutral, standing or idling position. Flange 26 is moved from the full line position to the forward or left hand dot and dash position shown in Figure 1.

Referring to Figures 1, 2 and 3 and especially to Figure 1, control valve 11 consists of a housing 49 secured to floor board 20 by means of a threaded end 50 and a nut 51. The housing is provided with three chambers 52, 53 and 54 interconnected by means of port 55 having a valve seat 56 and port 57 having a valve seat 58. A port 60 connects chamber 52 with a conduit 61. A port 62 connects chamber 53 with a conduit 63. A lever 65 pivotally connected at 66 to housing 49 has a disk 67 rotatably mounted at 68. Internally, threaded end 50 is provided with a bearing 70.

A piston 71 pivotally connected at 72 to lever 65 on one end is slidably mounted in bearing 70 on its opposite end. Piston 71 is provided with a valve flange 73 and a shoulder 74. A collar 75 is slidably mounted on piston 71 and has a washer of packing material 76 fixed to one side. A plug 77 is secured in one end of chamber 54 by means of a recess 78 in housing 49 and a snap ring 80. A spring 81 is interposed between collar 75 and plug 77. A clearance hole 82 in plug 77 permits piston 71 to slide therethrough. A port 83 in housing 49 connects chamber 54 with the atmosphere. An air filter 84 is fixed in port 83. It consists of two plates 85 and 86 each provided with a plurality of orifices. Wire mesh 87 is placed between plates 85, 86 which are secured in housing 49.

Power unit 12 consists of a housing 90 having a hydraulic cylinder 92 and a secondary vacuum chamber 91.

[PX 8, p 7]

Housing 90 is provided with a rim 93. A cover 94 is secured to rim 93. Housing 90 is provided with a boss 96 having a bearing 97. A circular groove 98 located in bearing 97 retains an O ring 99. A piston rod 100 provided with a flange 101, an enlarge portion 102 and a thread 103 is slidably mounted in bearing 97 and passes through O ring 99. A washer 104 abuts flange 101. A disk 105 is located adjacent washer 104. A piston ring 107 circular and L-shaped slidingly engages the internal diameter 106 of cover 94. The base of the L abuts disk 105. A second disk 108 abuts one surface of the L. A cap 110 provided with a recess 111 bears against second disk 110. An O ring 112 lies in recess 111 and engages the leg of the L of pisten ring 107. A nut 113 engages threads 103 forcing cap 110, second disk 108, piston ring 107, disk 105 and washer 104 together, as a unit, against flange 101. Cover 94 is provided with an indented portion which acts as a bumper for the end 114 of piston 100, to provide an atmospheric chamber 115 between cap 110 and the end of cover 94. A vacuum chamber 95 is formed between disk 105 and housing 90. A spring 116 is interposed between disk 105 and housing 90. Conduit 63 connects atmospheric chamber 115 with port 62 and chamber 53.

Housing 90 is provided with two passageways 117 and 118 leading from vacuum chamber 95 to secondary vacuum chamber 91. A cavity 120 having a conduit 124 leading to vacuum chamber 95 is provided in housing 90. A thread 121 is provided in the outboard end of cavity 120. A check valve 301 is located in cavity 120. It consists of a split ring 122, placed in a recess in housing 90 approximately midway of the depth of cavity 120, retaining a detent consisting of a ball 123 having a spring 123A interposed between split ring 122 and ball 123, whereby, ball 123 is normally held against the intersection of conduit 124 with cavity 120. Split ring 122 is provided with a port 125.

Housing 90 is provided with threads 126. A cap 127 is secured to threads 126 forming hydraulic cylinder 92. A hydraulic piston 130 is provided with a passageway 128. A slot 128A, in piston 130, is provided with packing material 131 and is slidably mounted in hydraulic cylinder 92. Hydraulic piston 130 is secured to piston rod 100 by means of pin 133. A yoke 132 is provided with a flat projection which engages passageway 128. That flat projection is smaller than passageway 128 thereby allowing a conduit to exist for the passage of hydraulic fluid. A spring 132A is interposed between the end of yoke 132 and the base of piston 130. The end of piston rod 100 retains the yoke 132 in slot 128. The yoke 132 is therefore slidably mounted in passageway 128 and is held in one position by means of spring 132A. A cap 134 provided with a port 135 is fixed to hydraulic piston 130 by means of snap retaining ring 129. A detent consisting of a ball 136 and a spring 137 interposed between ball 136 and cap 134 engages passageway 128. Fluid pressure forces ball 136 against the tension of spring 137 away from passageway 128. A pipe thread 140 located in housing 90 is connected to hydraulic cylinder 92 by means of conduit 141. A plug 140A is secured in thread 140. Power unit 12 is manufactured by the Bendix Aviation Corporation, Brake Division, South Bend, Indiana and is a Bendix Hydrovac, Model 500.

A cover 142 provided with a third vacuum chamber 143 and a outlet 144 is secured as by means of screws 446 to housing 90 with a diaphragm 145 inserted between housing 90 and cover 142. A conduit 61 connects outlet 144 with port 60. Diaphragm 145 therefore separates secondary vacuum chamber 91 from third vacuum chamber 143. Diaphragm 145 is provided with a port 146. A cap 147 provided with a projection 148 and a plurality of ports 150 is riveted to diaphragm 145 by means of a plurality of rivets 151 which also secure a cup shaped washer 152 to diaphragm 145. A plug 153 provided with a cavity 154 is secured to housing 90 by means of threads 155. Projection 148 passes through an orifice in plug 153 to project into cavity 154 and engage a piston consisting of a disk 156 and packing material 157. Housing 90 is provided with a cavity 158 which form a continuation of cavity 154. A conduit 160 is blocked by a plug 161 fixed in housing 90 and is provided as a manufacturing expedient in order to produce conduit 162 which connects cavity 158 with hydraulic cylinder 92. A plug 163 is provided in conduit 162 to seal the end of conduit 162. Cover 142 is provided with a hollow projection 165 having oppositely disposed ribs 166 and 167 which form a passageway 168. The hollow 170 in projection 165 forms part of a hollow core 171 in cover 142. An air filter consisting of a disk 172 having a plurality of orifices 172A and a disk 173 having a plurality of orifices 173A with filter material 174 between said disks 172 and 173 is fixed in hollow core 171 by means of split rings 175 and 176. A piston consisting of a shank 177 having oppositely disposed disk ends 180 and 181 is mounted to reciprocate in passageway 168. A spring 182 is interposed between cup shaped washer 152 and the base of rib 166. A spring 183 is interposed between disk 180 and disk 173.

Referring to Figure 1. The vacuum from engine manifold 17 manifests itself through conduit 300 to provide a vacuum in vacuum chamber 95 through check valve 301 by sucking or drawing ball 123 against the tension of spring 123A away from the intersection of conduit 124 with cavity 120. Thus a vacuum is created in vacuum chamber 95.

Referring to Figure 2. At this point in the operation of the motor vehicle, the accelerator pedal 21 is in the full line position, shown in Figure 2. Disk 67 and lever 65 are also in the full line position. Valve flange 73 is positioned in valve seat 56 blocking port 55. Piston 260 of power unit 12 is in the operative position shown in Figure 2 whereby hydraulic piston 130 forces hydraulic fluid out of hydraulic cylinder 92 through connection 302 leading to the brake cylinders on the wheels of the motor vehicle.

Referring to Figure 1, the vacuum in vacuum chamber 95 manifests itself through passageways 117, 118, secondary vacuum chamber 91, ports 150, port 146, third vacuum chamber 143, outlet 144, conduit 61, port 60 and chamber 52 where the vacuum is trapped by flange valve 73 engaging valve seat 56. Atmospheric pressure passes (Figure 2) through air filter 84 into chamber 54, port 57, chamber 53, port 62, conduit 63, and into atmospheric chamber 115 whereby said atmospheric pressure pushes piston 260 against the tension of spring 116 so that piston 260 moves forwardly from the position shown in Figure 1 to the position shown in Figure 2. Hydraulic piston 130 is thereby moved forwardly to force hydraulic fluid out of hydraulic cylinder 92 as previously described.

As the accelerator 21 is depressed to the full line position, refer to Figure 1, flange 26 is pushed forwardly from the right hand dot and dash line position to the full line position. Spring 81 acting against collar 75 and shoulder 74, forces piston 71 rearwardly to unseat valve flange 73 (as shown). Lever 65 is thereby moved to the neutral position shown in full lines in Figure 1 from the dot and dash line position. Continued depression of accelerator 21 from the full line position to the lower dot and dash position causes accelerator rod 25 to move from the neutral position illustrated in full lines to a position where head 380 engages surface 380A to move control sleeve forward whereby the carburetor admits more gasoline to the cylinder to speed up the engine. Simultaneously, atmospheric pressure is shut off from chamber 53 through the seating of packing material 76 engaging valve seat 58. Spring 116 forces piston 260 rearwardly.

The rearward action of piston 260 pulling hydraulic piston 130, releases the hydraulic fluid pressure on the brake cylinders to release the brakes. Further depression of accelerator pedal 21 causes the motor vehicles to travel at any speed in the conventional manner. The brakes are inoperative.

As the foot is released from depressing accelerator pedal 21, flange 26 moves from the left hand dot and dash position as shown in Figure 1 to the full line position. Flange 26 contacts disk 67. As the foot is further released from accelerator pedal 21, flange 26 moves disk 67 toward the right hand dot and dash position. This movement actuates piston 71 to cause piston 260 through hydraulic piston 130 to apply the motor vehicle brakes.

The structure and operation of the present invention with the minimum number of parts has just been described. In the event the engine stalls and also to provide a braking force for a short period of time after the ignition switch is turned off; a reserve vacuum supply is provided in conjunction with an emergency valve 14.

Referring to Figures 2 and 5 wherein is shown an emergency valve 14 consisting of a body 210 formed to provide a central passageway consisting of a chamber 211 and a core 212. Two pairs of oppositely disposed conduits 213, 214, 215, 216 are located in body 210. A cap 220 is provided with a conduit 221. A diaphragm 230 is secured between cap 220 and body 210 providing a chamber 228. A spring 231 is interposed between a washer 307 and cap 229. A piston 232 is provided with two disks 261 and 262 which slidably engage core 212. Disk 261 blocks conduits 213, 214. Disk 262 blocks conduits 215, 216. One end of piston 232 is secured to diaphragm 230 by means of a pin 305 which pivotally connects the end of piston 232 to the yoke of a rivet 306 which secures washer 307 to diaphragm 230. Two electrical contacts 233 and 234 are secured in body 210 by means of insulated bushings 308 and 309. The end of piston 232 is reduced to provide a surface 315 upon which a disk 310 electrically conductive, may slide. A washer 311 is positioned against the shoulder formed by the reduced portion 315 with the main body of piston 232. A second washer 312 separated from disk 310 by means of an insulated washer 313 is slidably mounted on the reduced portion 315. A spring 316 is interposed between washers 311 and 312.

A vacuum supply tank 16 is provided with a conduit 317 connecting the manifold 17 with vacuum supply tank 16. Conduit 214 connects vacuum supply tank 16 with core 212. Conduit 221 connects chamber 228 with manifold 17. Conduit 300 as previously described in conjunction with the mechanism shown in Figure 1 is made to connect conduits 213 and 215 with check valve 301 instead of connecting check valve 301 with manifold 17. Conduit 216 connects core 212 with manifold 17. Electrical contacts 233 and 234 are not put to use in the arrangement shown in Figure 2. Their purpose will appear in conjunction with the mechanism shown in Figure 3.

When the engine of the motor vehicle is functioning manifold 17 creates a vacuum in vacuum supply tank 16. The supply in vacuum supply tank 16 is approximately six times the volume of vacuum chamber 95 of power unit 12.

When the operative turns the ignition key off after the motor vehicle has been brought to a stop and the operatives foot has been removed from the accelerator pedal 21, the control valve 11 and power unit 12 will have their moving parts in the position shown in Figure 2. The brakes are applied. In this position disk 262 blocks conduits 215, 216. Disk 261 allows the vacuum from vacuum supply tank 16 to pass through conduit 214 into core 212, conduit 213, conduit 300, past check valve 301 into vacuum chamber 95. The vacuum then causes the functions previously described.

As soon as the engine is stopped back pressure from the manifold 17 will pass through conduit 221 into chamber 228 assisting spring 231 in forcing piston 232 and disks 261 and 262 into the position whereby disk 261 allows the vacuum to pass from conduit 213, core 212 to conduit 213. The vacuum in core 212 and chamber 211 aids spring 231 in flexing diaphragm 230 to move piston 232.

When the engine is running and a vacuum is created in manifold 17, said vacuum will pass through conduit 221 into chamber 228 and thereby such diaphragm 230 against the tension of spring 231, positioning piston 232 and disks 261, 262 so that disk 261 blocks conduits 213, 214 and disk 262 allows the vacuum from manifold 17 to pass through conduit 216, core 212, conduit 215, conduit 300 to vacuum chamber 95.

The mechanism shown in Figure 3 is a further modification of the mechanism shown in Figures 1 and 2. Additional features of safety are provided such as an anti-skid unit 13 used in conjunction with a solenoid 19, a solenoid check valve 15, a check valve 7 and an electrical system under control of the ignition switch.

Referring to Figures 3 and 6 wherein is shown the anti-skid unit 13 consisting of a housing 320 forming a cavity 321, a gear box 322 and provided with a conduit 323. An inertia governor generally indicated at 324 and located in cavity 321 consists of two oppositely disposed arms 325 and 326. Counterweight 327 is pivotally connected to arms 325 and 326 by means of links 328 and 329. Counterweight 330 is pivotally connected to arms 325 and 326 by means of links 331 and 332. A spring 333 is tensioned and connected on opposite ends to counterweights 327 and 330.

A tubular body 333 is divided into two sections 334 and 335 by means of a separator 336 fixed in said body 333. Said body 333 is secured to housing 320. A wire coil 337 is located in section 335. By means of insulated bushings 338 and 339 wire coil 337 provides wire connections 349, 341. A plunger 342 is mounted for reciprocation through electrical induction of coil 337 in separator 336. A spring 343 is interposed between separator 336 and a shoulder on plunger 342. Conduit 323 is shaped to form a chamber 344. Plunger 342 passes through an opening in housing 320 to extend into chamber 344. A piston rod 345 slidably mounted in housing 320 is secured on one end to arm 325. A block 346 is pivotally mounted to the other end of piston rod 345 by means of a pin 347. Block 346 is slidably mounted in chamber 344. A spring 348 is interposed between housing 320 and block 346. The end of plunger 342 engages block 346. Block 346 moves into and out of conduit 323 as will presently appear. Arm 326 is provided with an extension 350 having a thread 351. See Figure 9. Arm 326 is rotatably mounted in a bushing 352 fixed in housing 320. A pinion gear 353 is rotatably mounted on threads 351. Housing 320 is provided with a recess 354 adapted to accommodate a collar 355 fixed to the end of extension 350. A shaft 349 connected to the speedometer cable of the motor vehicle is rotatably mounted in gear box 322. A gear 357 secured to shaft 349 meshes with pinion 353. A plunger 358 fabricated from insulated material is slidably mounted in gear box 322. An electrical grounded spring contact 360 is fixed to gear box 322. Plunger 358 engages contact 360. A cover 361 is secured to gear box 322. A wire contact 362 is secured in cover 361 by means of insulated bushing 363. Spring contact 360 under the influence of plunger 358 engages contact 362.

Referring to Figures 7 and 7 wherein is shown a solenoid check valve generally indicated at 15. It comprises a housing 190 having an axial core 191, provided with a bore recess 192 and a valve seat 193. Bores 194, 195 are located on opposite sides of housing 190 and connect with axial core 191 on opposite sides of valve seat 193. A separator 196 is fixed in axial core 191. A wire coil 197 is located in axial core 191 on one side of separator 196. A plate 198 fixed to housing 190 is provided with insulated bushings 199 and 200 which accommodate ends 201 and 202 of wire coil 197. A plunger 203 is reciprocately mounted in separator 196 and coil 197. It is provided with a valve head 204 and an

extension 205 which slidably engages base recess 196. A spring 206 is interposed between valve head 204 and separator 196. Electrical induction in coil 197 on plunger 203 overcomes the tension of spring 206 causing valve head 204 to engage valve seat 195. A conduit 207 connects conduit 193 with fitting 302. A conduit 208 connects conduit 194 with the hydraulic brake cylinders on the wheels.

Referring to Figures 3 and 8 wherein is shown a check valve 7. Conduit 317 leading from manifold 17 is formed into a cup 365. A conduit 366 connected on one end to vacuum supply tank 16 is formed into a plug 367 having a cavity 368. Plug 367 engages cup 365 in a fluid tight relationship. A detent consisting of a ball 370 and a spring 371 is located in cavity 368 with the ball 370 engaging and disengaging the hollow core of conduit 366.

Referring to Figure 3 and the linkage connecting the accelerator pedal 21 to the carburetor. The carburetor rod 372 leading from the motor vehicle carburetor is threaded on end 373. A connector 374 is provided with male threads 375 on one end and female threads adapted to receive threads 373 on the other end. A lock nut 376 secures carburetor rod 372 with connector 374. A control sleeve 24 is provided axially with a hollow sleeve 377 and female threads adapted to accommodate male threads 375. A lock nut 378 secures male threads 375 in control sleeve 24. A stud 27 provided with threads which engage threads in intermediate link 25 secures flange 26 between stud 27 and link 25. Stud 27 is provided with a head 380 which slidably engages hollow sleeve 377.

Referring to Figure 4 wherein is shown a schematic wiring diagram: A motor vehicle storage battery is generally indicated by reference numeral 5. "G" represents the grounded side of the battery. A conduit 390 leads from the battery to the ignition switch 6. A conduit 391 leads from the ignition switch 6 to conduit 341 of the anti-skid unit 13. Conduit 340 from coil 337 leads to electrical contact 362. Spring contact 360 when closed energized coil 337. Spring contact 360 is grounded at G2. Conduit 391 also connects ignition switch 6 with conduit 234 of emergency valve 14. Conduit 233 connects with conduit 202 of the coil 197 in solenoid check valve 15. Conduit 201 of coil 197 is grounded at G3. When conduits 233 and 234 of emergency valve 14 are connected through disk 310, the circuit leading to the solenoid check valve 15 is closed actuating plunger 204 and thereby maintain the hydraulic brake fluid in the brake cylinders; retaining the brakes operative.

Operation

The turning of the ignition switch starting the engine creates a vacuum in engine manifold 17. The operation of vacuum supply tank 16, emergency valve 14, control valve 11 and power unit 12 has been previously described.

The conventional hand or emergency brake will be employed when the motor vehicle is parked with the engine inoperative.

The present accelerator brake control is operative with a clutch type transmission. However, the full advantage of the present invention is lost if an operative must throw out the clutch to shift to cause movement of the motor vehicle. An automatic type transmission is preferably used in conjunction with the present invention.

Referring to Figures 3 and 4, the turning of the ignition switch 6 closes the electrical circuit. Electrical current from the battery 5 will travel through conduit 391 to coil 337 and contact 234. Coil 337 will become energized when contact 360 engages contact 362 as will presently appear. Contacts 233 and 234 when engaged by disk 310 will energize coil 197 as will presently appear.

The emergency brake is released and the operative's foot engages and depresses accelerator pedal 21 causing intermediate link 25 to move flange 26 from the right hand dot and dash position to the full line position as shown in Figure 3. The movement of lever 65 releases the hydraulic brakes through control valve 11 and power unit 12 as previously described. Simultaneously head 380 sliding in chamber 377 of control sleeve 24 will move through a "neutral" zone. That is a zone or area where depressing the accelerator pedal 21 produces an ineffectural travel before engaging the connector 374 to actuate the carburetor rod 372 to feed more gasoline to the cylinders to speed up the engine. This area of travel is a time interval to allow the brakes of the motor vehicle to be released.

An observation is made that this time interval of extremely short duration replaces the time interval formerly required by the operative to shift a foot from the accelerator pedal 21 to the brake pedal. One of the outstanding accomplishments of the present invention is to reduce the time lag between removing an operative's foot from the accelerator to applying the brake.

This neutral zone also allows the operative to permit the engine to cruise without applying the brake because head 380 in chamber 377 must return to the "brake" zone before flange 26 engages disk 67 to actuate lever 65 and piston 71 to set in motion the mechanism to apply the vehicle's brakes.

Referring to Figure 3. The motor car is now travelling at a desired speed in the conventional manner. The brakes are inoperative. If the motor vehicle is to be stopped in the normal manner, the foot is gradually lifted from accelerator pedal 21 setting in motion the mechanism of control valve 11, power unit 12, emergency valve 14 and vacuum supply tank 16 as previously described.

If the operative's foot is jerked free of accelerator pedal 21, as with a sudden jolt, then the mechanism of the anti-skid device 13 is set in motion to prevent the wheels of the motor vehicle from locking under the sudden impact of the hydraulic fluid. The sudden withdrawal of the foot from accelerator pedal 21 causes piston 71 to open chamber 53 to atmospheric pressure as previously described. Atmospheric pressure travels through conduit 63 to atmospheric chamber 115. A vacuum having been created in vacuum chamber 95. The sudden denial of gasoline to the engine causes the engine to slow down rapidly causing the motor vehicle to slow up or check itself. Speedometer cable 349 is attached on one end to the drive shaft of the motor vehicle. Therefore the rotation of the speedometer cable 349 drops rapidly causing gear 357 to suddenly slow its rotation. Governor 324 having the rotation or inertia of the original speed and the governor 324 inherently trying to maintain that speed, rotates extension 350 at that speed. Gear 357 engaging pinion 353 attempts to hold back the rotation of pinion 353. Pinion 353 is forced to yield because the pinion drives the governor and the governor is rotating at a greater speed. Pinion 353 walks on threads 351 toward insulated rod 358 forcing rod 358 against contact 360 and thereby closes the gap between contacts 360 and 362 closing the electric circuit from battery 5 whereby coil 337 is energized actuating piston 342 by drawing it into coil 337, and away from plug 346. Governor 324 begins to slow up. Springs 356 and 348 force the links 327 and 330 toward each other thereby forcing plug 346 out of the position whereby it blocks conduit 323. This action allows atmospheric pressure from conduit 63 to pass through conduit 323 to chamber 143. Disk 181 is away from cup shaped washer 152. Atmospheric pressure therefore enters vacuum chamber 95 from chamber 143 to release piston 260 to the influence of spring 116 by providing atmospheric pressure on both sides of piston 260. Thereby releasing the brakes through the movement of hydraulic piston 130 in hydraulic cylinder 92. This action takes place instantaneously. As the governor 324 slows down, pinion 353 through the slower rotation of gear 357, rotates at a slower rate of speed than ex-

tension 350. Pinion 353 rotates on threads 351 to reposition itself away from insulated rod 358 to normal position. The contacts 360 and 362 separate to break the electrical circuit. Coil 337 is deenergized freeing piston 342 to the action of spring 343 whereby piston 342 engages head 346 in position to block conduit 323. Thereby shutting off atmospheric pressure to vacuum chamber 95 and permitting vacuum to again enter vacuum chamber 95 so that piston 260 can again be actuated to apply the brakes.

Assuming the motor vehicle is traveling at a normal speed in a normal way on a level grade or down hill. Suddenly the engine stops due to engine failure. Check valve 7 instantly traps the vacuum in vacuum supply tank 16 because the vacuum in the engine manifold 17 is lost. Atmospheric pressure replaces the vacuum in manifold 17. Atmospheric pressure from manifold 17 passes through conduit 221 to chamber 228 allowing spring 231 to actuate diaphragm 230 to force disk 310 to close the circuit between contacts 233, 234 to energize coil 197, thereby drawing piston 203 into coil 197. Valve plug 204 engages valve seat 195 to block communication between conduits 207, 208. Disk 262 moves into position between ports 215, 216. Disk 261 moves out of blocking position in front of ports 213, 214 to permit free communication from vacuum supply tank 16 through conduit 214 to core 212, port 213, conduit 300 to vacuum chamber 95. Thereby maintaining the vacuum in vacuum chamber 95.

The operative's foot will be raised or removed from accelerator pedal 21 to apply the brake. The brakes will be applied as previously described.

Hydraulic piston 130 moves forwardly in hydraulic cylinder 92 forcing hydraulic fluid through conduit 207, into chamber 191 and thereby lifting valve plug 204 away from valve seat 195 against the pull of coil 197 to allow hydraulic fluid to enter conduit 208 leading to the brake cylinders. The moment the pressure is released in hydraulic cylinder 92 coil 197 pulls valve plug 204 into engagement with valve seat 195. Thereby trapping the fluid in the brake cylinders. Reactivation of the engine will create a vacuum in manifold 17 and through conduit 221 a vacuum will be created in chamber 228. Diaphragm 230 is sucked rearwardly against the tension of spring 231 thereby pulling disk 310 away from contacts 233 and 234 interrupting the circuit to coil 197. Spring 206 raises valve plug 204 away from valve seat 195 permitting the pressure of hydraulic fluid to be released from the brake cylinders.

It is contemplated that the present invention will replace the conventional foot brake pedal on a motor car. However, provision is made to allow the present invention to be used with a convention footbrake pedal. Flange 26 is removed to render inoperative control valve 11 with the various parts as shown in Figure 1 wherein spring 81 maintains flange valve 73 unseated from valve seat 56 and washer 76 seated against valve seat 58. When the foot brake pedal is applied hydraulic fluid in the conduit leading from the foot brake pedal to the threads 140 in housing 90 will pass into conduit 141 and into chamber 92 from where it will pass into conduit 162 and cavity 158. The fluid in cavity 158 will actuate disk 156 thereby forcing projection 148 and diaphragm 145 into position to engage disk 181 (see Figure 2) whereby piston 177 moves disk 180 away from rib 167 against the tension of spring 183 to allow atmospheric pressure to pass through port 171, cavity 170, chamber 143, outlet 144, conduit 61, port 60, chamber 52, port 55, chamber 53, conduit 63 into atmospheric chamber 115 where piston 260 applies the braking force as previously described. Cap shaped washer 152 engages disk 181 so that port 146 is rendered ineffectual.

It will be noted that disk 180 is held against rib 167 by spring 183 preventing atmospheric pressure from passing into passageway 168 from hollow core 171 when the accelerator 18 is used without a brake pedal and in place of the brake pedal.

Spring 81 holds collar 75 and washer 76 against valve seat 58 preventing atmospheric pressure from passing from port 83 to chamber 53 when the conventional brake pedal is used with the present mechanism. The accelerator 18 performing its normal function.

Referring to Figure 10 wherein is shown a schematic perspective view of a motor vehicle, by way of example but not of limitation, a hydraulic automobile brake system having a master hydraulic cylinder 400, brake members 401, 401A, 401B, 401C, expandable against brake bands (not shown) on wheels indicated by dot and dash lines 399, 399A etc., by any conventional means, such as hydraulic cylinders 402, 402A, 402B, 402C.

Fluid conduit 403 connects hydraulic cylinders 402, 402C. Fluid conduit 404 connects hydraulic cylinders 402A, 402B. A common fluid conduit 207 connects fitting 302 with fluid conduits 403 and 404. A fluid conduit 406 connects pipe thread 140 in power unit 12 with master cylinder 400.

A conventional brake pedal 407 is connected to the conventional master hydraulic cylinder 400 of the usual type, so that depression of the brake pedal (to the left, as illustrated in Figure 10) against the tension of a customary frame-anchored coil spring 408 as about a pivot 410, a plunger 411 pivotally secured to the pedal lever applies pressure to the brake fluid within the master cylinder and through fluid conduit 406 to conduit 141 as previously described, it being understood that use of the conventional brake pedal 407 necessitates the removal of flange 26 to render inoperative brake control through accelerator pedal 21. Conversely, brake pedal 407, master hydraulic cylinder 400 are contemplated to be eliminated by the present invention with the substitution of plug 140A for conduit 406.

A conventional internal combustion engine for the motor vehicles is indicated at 415 and is provided with a conventional carburetor indicated at 416 and a manifold 17. A linkage generally indicated at 417 connects carburetor rod 372 to carburetor 416.

Referring to Figure 11 wherein is shown a modified form of anti-skid device 13 and a modified form of powder unit 12 with the speedometer cable 349 connection; the conventional combustion engine indicated at 415 is provided with a conventional torque converter or fluid drive transmission indicated at 420 having the conventional drive shaft indicated at 421 and a conventional speedometer cable 349 connected to drive shaft 421.

The power unit 12 illustrated in Figure 11 provides for conduit 61 connecting chamber 52 through outlet 144A directly to vacuum chamber 95 instead of connecting chamber 52 with outlet 144, third vacuum chamber 143, secondary vacuum chamber 91 and vacuum chamber 95. Third vacuum chamber 143 and secondary vacuum chamber 91 along with the mechanism located in these chambers may be eliminated if it is contemplated to not provide for use of conventional brake pedal 407.

Inertia governor 324 in the form shown in Figure 11 differs from Figures 3 and 6 in that the electrical unit is dispensed with in this modification.

Having illustrated and described preferred embodiments of the present invention, by way of example, but realizing that structural changes could be made and other examples given without departing from either the spirit or scope of this invention.

What I claim is:

1. An accelerator brake control consisting of an engine manifold provided with a vacuum, a control valve and a power unit comprising a housing having a chamber, a spring loaded piston separating said chamber into a vacuum side and an atmospheric pressure side, said control valve selectively permitting passage of atmospheric air into the pressure side of said chamber and selectively permitting a vacuum condition to be present in said

atmospheric pressure side, a conduit provided between said engine manifold and said vacuum side whereby in one selected position of said control valve a vacuum is present in said vacuum side and in said atmospheric pressure side to condition said piston to the influence of said spring and in another selected position of said control valve a vacuum is present in said vacuum side and atmospheric pressure is present in said atmospheric pressure side whereby said piston is moved in said chamber against the force of said spring a second valve is provided, an inertia governor is operatively connected to said second valve for actuating said second valve to admit atmospheric air to the vacuum side of said chamber when said inertia governor is in operative position and said control valve has selectively permitted a vacuum condition to be present in said vacuum side of said chamber.

2. An accelerator brake control for a motor vehicle having a plurality of wheels and consisting of an engine manifold provided with a vacuum, an accelerator, a control valve and a power unit comprising a housing having a chamber, a spring loaded piston separating said chamber into a vacuum side and an atmospheric pressure side, said control valve selectively permitting passage of atmospheric air into said atmospheric pressure side of said chamber and selectively permitting a vacuum condition to be present in said atmospheric pressure side of said chamber, linkage means operated by the accelerator for selectively actuating said control valve, a conduit between said engine manifold and said vacuum side whereby in one selected position of said accelerator a vacuum is present in said vacuum side and through said control valve on said atmospheric pressure side to condition said piston to the influence of said spring and in another selected position of said accelerator a vacuum is present in said vacuum side and through said control valve atmospheric pressure is present in said atmospheric pressure side whereby said piston is moved in said chamber against the force of said spring, a second valve is provided, an inertia governor is provided controlling said second valve, a second means operatively connected to said inertia governor is provided for actuating said second valve to admit atmospheric air to the vacuum side of said chamber, a vacuum supply tank, a check valve and an emergency valve are positioned in said conduit between said engine manifold and said vacuum side whereby said engine manifold in one condition creates a vacuum in said vacuum supply tank by passing through said check valve, while at the same time said engine manifold maintains said emergency valve in position to block escape of said vacuum from said vacuum supply tank and said engine manifold in another condition when atmospheric air is present in said engine manifold permits said emergency valve to move to a second position whereby said emergency valve blocks out the influence of said engine manifold on said vacuum side through said conduit and unblocks the escape of vacuum from said vacuum supply tank to said vacuum side to substitute the vacuum from said vacuum supply tank for the vacuum from said engine manifold, said check valve blocking the influence of atmospheric air in said engine manifold upon said vacuum supply tank.

3. A mechanism as defined in claim 2 being further characterized in that a first hydraulic cylinder is provided, a second piston slidably mounted in said hydraulic cylinder is provided and connected to said first mentioned piston, a hydraulic cylinder provided on each of said plurality of wheels, a conduit connecting each of said hydraulic cylinders on said plurality of wheels with said first hydraulic cylinder, the movement of said first mentioned piston moving said second piston to actuate said hydraulic cylinders on each of said plurality of wheels.

4. An accelerator brake control consisting of an engine manifold having a vacuum, an accelerator, a control valve and a power unit comprising a housing having a chamber, a vacuum operated first piston slidably mounted in said chamber and dividing said chamber into a vacuum chamber and an atmospheric chamber, a first conduit connecting said vacuum in said engine manifold with said vacuum chamber, a hydraulic cylinder formed in said housing, a piston rod forming part of said first piston, a hydraulic piston slidably mounted in said hydraulic cylinder connected to said piston rod, a spring in said vacuum chamber urging said first piston in a direction to reduce the area of the atmospheric chamber, a control valve provided with a first chamber, a second chamber and a third chamber, a port on one side of said first chamber exposed to the atmosphere, a second port between said first chamber and said second chamber, a third port between said second chamber and said third chamber, a third piston operably connected to said accelerator and reciprocally mounted in said control valve and provided with a first valve seat engageable and disengageable with said third port, a second seat on said third piston engageable with said second port when said first valve seat is disengaged from said third port and disengageable from said second port when said first valve seat engages said third port, a second conduit uniting said second chamber with said atmospheric chamber and a third conduit uniting said third chamber with said vacuum chamber.

5. An accelerator brake control consisting of a manifold producing a vacuum, an accelerator, a control valve and a power unit comprising a housing having a chamber, a vacuum operated first piston having a piston rod, slidably mounted in said chamber and dividing said chamber into a vacuum chamber and an atmospheric chamber, a first conduit connecting said manifold with said vacuum chamber, a bumper in said housing engageable with said first piston to provide a minimum area for said atmospheric chamber, a hydraulic cylinder having an outlet formed in said housing, a hydraulic piston slidably mounted in said hydraulic cylinder connected to said piston rod, a spring located in said vacuum chamber acting on said first piston, a control valve provided with a first chamber, a second chamber and a third chamber, a first port on one side of said first chamber leading to the atmosphere, a second port between said first chamber and said second chamber, a third port between said second chamber and said third chamber, a third piston operably connected to said accelerator and reciprocally mounted in said control valve and provided with a first valve seat engageable and disengageable with said third port, a second valve seat on said third piston engageable and disengageable with said second port, a second conduit uniting said second chamber with said atmospheric chamber and a third conduit uniting said third chamber with said vacuum chamber, said accelerator in one position actuating said third piston to engage said first valve seat with said third port and to disengage said second valve seat from said second port to allow atmospheric pressure entering said first port and first chamber to pass through said second port, second chamber, second conduit and enter said atmospheric chamber to actuate said first piston against the resistance of said first spring whereby said hydraulic piston slidably moves in said hydraulic cylinder, vacuum from said manifold passing through said first conduit to said vacuum chamber and through said third conduit to said third chamber, said accelerator in a second position actuating said third piston to disengage said first valve seat from said third port and to engage said second valve seat with said second port to allow vacuum passing from said manifold through said first conduit, vacuum chamber, third conduit, third chamber, third port, second chamber, second conduit to said atmospheric chamber where a vacuum is created to allow said first spring to position said first piston against said bumper and hold said hydraulic piston in a preselected position in said hydraulic cylinder.

6. An accelerator brake control consisting of an engine manifold having a vacuum, an accelerator, a control valve, a vacuum supply tank, an emergency valve and a power unit comprising a housing having a chamber, a vacuum operated first piston slidably mounted in said chamber and dividing said chamber into a vacuum chamber and an atmospheric chamber, a hydraulic cylinder formed in said housing, a piston rod forming part of said first piston, a hydraulic piston slidably mounted in said hydraulic cylinder connected to said piston rod, a spring in said vacuum chamber acting on said first piston, a control valve provided with a first chamber, a second chamber and a third chamber, a port on one side of said first chamber leading to the atmosphere, a second port between said first chamber and said second chamber, a third port between said second chamber and said third chamber, a third piston operably connected to said accelerator and reciprocally mounted in said control valve and provided with a first valve seat engageable and disengageable with said third port, a second seat on said third piston engageable and disengageable with said second port, a second conduit uniting said second chamber with said atmospheric chamber, a third conduit uniting said third chamber with said vacuum chamber, said emergency valve comprising a body having a passageway consisting of a fifth chamber and a core, a cap having a fourth conduit, a diaphragm, said cap securing said diaphragm to said body at said fifth chamber to provide a sixth chamber, a spring positioned between said cap and said diaphragm, a fourth piston having a first disk and a second disk slidably mounted in said core and fixed on one end to said diaphragm, a fifth conduit and an oppositely disposed sixth conduit located in said body, a seventh conduit and an oppositely disposed eighth conduit located in said body, said sixth and eighth conduit interconnecting to provide a first conduit connecting said core with said vacuum chamber, said fourth conduit connecting said sixth chamber with said engine manifold, said fifth conduit connecting said core with said engine manifold, a ninth conduit connecting said engine manifold with said vacuum supply tank, and seventh conduit connecting said core with said vacuum supply tank, said seventh conduit connecting said core with said vacuum supply tank, said first disk engaging and disengaging said fifth and sixth conduits, said second disk engaging said seventh and eighth conduits when said first disk disengages said fifth and sixth conduit and said second disk disengaging said seventh and eighth conduits when said first disk disengages said fifth and sixth conduits.

7. An accelerator brake control consisting of an engine manifold provided with a vacuum, an accelerator, a control valve, a vacuum supply tank, an emergency valve and a power unit, linkage means operated by the accelerator for selectively actuating said control valve said power unit comprising a housing having a chamber, a spring loaded piston separating said chamber into a vacuum side and an atmospheric pressure side, said control valve under control of said accelerator selectively permitting passage of atmospheric air into the pressure side of said chamber and selectively permitting a vacuum condition to be present in said atmospheric pressure side, a connection between said engine manifold and said vacuum supply tank, creating a vacuum in said vacuum supply tank, said emergency valve having a diaphragm operated piston, a second connection between said engine manifold and one side of said diaphragm, a third connection between said engine manifold and the other side of said diaphragm, a fourth connection between said vacuum supply tank and the last mentioned side of said diaphragm, a fifth connection between said last mentioned side of said diaphragm and said vacuum side, in one selected position of said accelerator a vacuum from said engine manifold is present in said vacuum side and through said control valve vacuum from said vacuum side is present in said atmospheric pressure side to condition said piston to the influence of said spring, in another selected position of said accelerator a vacuum from said engine manifold is present in said vacuum side and through said control valve atmospheric air is present in said atmospheric pressure side whereby said piston is moved in said chamber against the force of the spring, the vacuum in said engine manifold acting on said diaphragm to permit the vacuum from the engine manifold to pass through said emergency valve to said vacuum side, the loss of vacuum in said engine manifold actuating said diaphragm operated piston to permit the vacuum from said vacuum supply tank to pass through said emergency valve to said vacuum side, said diaphragm actuated piston blocking the connection between said engine manifold and vacuum side when the vacuum in said engine manifold is lost.

8. An accelerator brake control consisting of an engine manifold provided with a vacuum, an accelerator, a control valve, an anti-skid unit provided with a first solenoid, a vacuum supply tank, a first check valve, an emergency valve, a solenoid check valve and a power unit, linkage means operated by the accelerator for selectively actuating said control valve, said power unit comprising a housing having a first chamber, a hydraulic cylinder, a secondary vacuum chamber, a third vacuum chamber, a spring loaded first piston separating said first chamber into a vacuum side and an atmospheric pressure side, a hydraulic piston slidably mounted in said hydraulic cylinder and operatively connected to said first piston, a first diaphragm having a first port interposed between said secondary vacuum chamber and said third vacuum chamber, a second port between said vacuum side and said secondary vacuum chamber, means operatively connected to said hydraulic cylinder to block said first port in said first diaphragm, said control valve under control of said accelerator selectively permitting passage through a first conduit of atmospheric air into said atmospheric pressure side of said chamber and selectively through said first conduit permitting a vacuum condition to be present in said atmospheric pressure side, a second conduit housing said first check valve between said engine manifold and said vacuum supply tank allowing a vacuum to pass from said manifold to said vacuum supply tank, said emergency valve having a diaphragm operated spring loaded second piston, a third conduit between said engine manifold and the spring loaded side of said diaphragm, a fourth conduit between said engine manifold and the side opposite said spring loaded side of said diaphragm, a fifth conduit between said vacuum supply tank and the side opposite said spring loaded side of said diaphragm, a second check valve, a sixth conduit housing said check valve connecting said side opposite said spring loaded side of said diaphragm and said vacuum side of said first piston, a speedometer cable, said anti-skid device consists of a housing forming a cavity, a gear box and a seventh conduit, a governor provided with two oppositely disposed arms is rotatively mounted in said cavity, a pinion mounted for transverse movement is rotatively mounted on one of said arms, a first switch is secured on said gear box and is operatively actuated by the transverse movement of said pinion, a gear operatively connected to said pinion is operatively connected to said speedometer cable to be rotated thereby, a block secured to the second of said oppositely disposed arms is slidably mounted to open and close said seventh conduit, a first solenoid secured to said housing is operatively connected to said block, said seventh conduit connecting said first conduit with said second conduit an eighth conduit connecting said hydraulic cylinder with said solenoid check valve, and electrical circuit having a source of electric power and a second switch operatively connects said first solenoid, said first switch and said solenoid check valve with said source of electric power and a ninth conduit connecting said control valve with said seventh conduit.

9. An accelerator brake control consisting of an engine manifold having a vacuum, an accelerator, a control valve and a power unit comprising a housing having a first chamber, a hydraulic cylinder, a secondary vacuum chamber, a third vacuum chamber, a vacuum operated spring loaded first piston slidably mounted in said first chamber and dividing said first chamber into a first vacuum chamber and an atmospheric chamber, a first conduit connecting said first vacuum chamber with said engine manifold, a hydraulic second piston slidably mounted in said hydraulic cylinder operatively connected to said first piston, linkage means operated by said accelerator for selectively actuating said control valve, a second conduit uniting said control valve with said atmospheric pressure chamber, a third conduit uniting said control valve with said third vacuum chamber, means between said third vacuum chamber and said secondary vacuum chamber under control of the fluid in said hydraulic cylinder opening and closing communication between said secondary vacuum chamber and said third vacuum chamber, a port between said secondary vacuum chamber and said first vacuum chamber, said control valve under control of said accelerator selectively permitting passage of atmospheric air into said atmospheric chamber whereby said spring loaded first piston is actuated against the tension of said spring to move said hydraulic piston in said hydraulic cylinder in one direction and said control valve under control of said accelerator selectively permitting a vacuum condition to be present in said atmospheric side to condition said spring loaded first piston to the influence of the force of said spring to actuate said spring loaded first piston to move said hydraulic piston in an opposite direction.

10. A control valve having a housing provided with a first chamber, a second chamber, a third chamber, a first port in said housing between said first chamber and the atmosphere, a second port in said housing between said first chamber and said second chamber, a first valve seat in the edge of said port facing said first chamber, a third port in said housing between said second port and said third port, a second valve seat in said port facing said third chamber, a fourth port in said housing exiting said second chamber, a fifth port in said housing exiting said third chamber, a piston mounted for reciprocating movement in said housing and provided with two valves adapted to alternately engage said first valve seat and said second valve seat, a lever pivotally mounted in said housing and pivotally connected to said piston, an accelerator pedal, linkage connecting said accelerator pedal and said lever consisting of an intermediate link, a flange, a stud provided with a head securing said flange to said intermediate link, a control sleeve provided with a hollow area, said head slidingly mounted in said hollow so as to push said control sleeve in one direction when in one end position of said hollow and to pull said control sleeve in an opposite direction when in the opposite end position of said hollow and a neutral or ineffectual operating position in said hollow when said head is intermediate the opposite ends of said hollow, linkage means connecting said accelerator pedal to said intermediate link and adjustable rod means fixed in said control sleeve, said flange engaging and disengaging said lever to actuate said piston to cause alternately one or the other of said two valves on said piston to engage and disengage the respective valve seats.

11. A governor controlled solenoid valve comprising a housing forming a cavity, a gear box, a conduit formed in said housing, a governor rotatively mounted in said cavity, consisting of two oppositely disposed arms, two oppositely disposed counterweights pivotally connected to opposite ends of said arms by means of links, resilient means suspended between and connected to said counterweights, a block slidably mounted and normally positioned in said conduit, operatively connected to one of said arms by means of a piston rod slidably mounted

in said housing, a tubular body fixed to said housing, a separator fixed in said tubular body and dividing said body into two cavities, a wire coil located in one of said cavities, a plunger reciprocatably mounted in said separator and said coil and abutting said block, a shoulder on said plunger, resilient means interposed between said separator and said shoulder, a pinion located in said gear box and mounted on the other of said oppositely disposed arms, to rotate said arm and to have limited lateral movement on said arm, a gear rotatively mounted in said gear box, means to drive said gear, an electrical switch secured to said gear box, means actuated by said lateral movement of said pinion to open and close said electrical switch, an electrical circuit connecting said switch with said wire coil, a sudden change in the speed of rotation of said gear effecting lateral movement of said pinion to close said electrical switch to energize said wire coil to withdraw said plunger from abutting position against said block to permit said resilient means to draw said counterweights toward each other to actuate said piston rod to move said block out of blocking position in said conduit.

12. An accelerator brake control consisting of an engine manifold normally under vacuum, a power unit, an accelerator, a first means for normally maintaining said power unit ineffective, a second means connecting said manifold with said power unit for opposing said first means for maintaining said power unit ineffective, a control valve movable to two positions, in one of which it seals said power unit against atmospheric pressure and in another position of which it admits said atmospheric pressure to overcome said first means for maintaining said power unit ineffective in conjunction with the vacuum from said engine manifold, a third means connecting said accelerator to said control valve to admit atmospheric pressure during one range of the travel of said accelerator and to close said control valve to atmospheric pressure during another range of travel of said accelerator, a second valve, an inertia governor, means to activate said inertia governor, said inertia governor operatively connected to said second valve whereby when said inertia governor is activated atmospheric pressure is admitted to overcome said second means to allow said first means to maintain said power unit ineffective so long as said inertia governor is activated and to allow said second means to oppose said first means when said inertia governor is de-activated.

13. An accelerator brake control consisting of an engine manifold normally under vacuum, a power unit, an accelerator, an inertia governor unit, a first means for normally maintaining said power unit ineffective, a second means connecting said engine manifold with said power unit, a control valve movable to an open position and a closed position, in closed position said control valve seals said power unit against atmospheric pressure, in open position it admits atmospheric pressure to overcome said first means for maintaining said power unit ineffective, in conjunction with the vacuum from said engine manifold, a third means connecting said accelerator to said control valve to move said control valve to open and closed positions, a conduit between said control valve and said power unit, a valve in said conduit under control of said inertia governor unit normally closed, a fourth means operable upon said inertia governor to actuate said valve to open position in said conduit to admit atmospheric pressure in opposition to the vacuum from said engine manifold which aids said atmospheric pressure in overcoming said first means which renders said power unit ineffective.

14. In an accelerator brake control for a motor vehicle the combination of having an engine manifold normally under vacuum, a power unit for applying brakes, an accelerator pedal, a first means for normally maintaining said power unit ineffective for applying the brakes, a second means connecting said manifold to said power unit

for opposing said first means for maintaining the power unit ineffective, a control valve movable to two positions, in one of which it seals said power unit against atmospheric pressure, and in the other of which it admits atmospheric pressure to overcome the said first means for maintaining the power unit ineffective in conjunction with the vacuum from said manifold, and a third means connecting said accelerator pedal to said control valve to admit atmospheric pressure during one range of its travel and to close said control valve during its movement through the driving range of said accelerator pedal.

15. An accelerator brake control consisting of an engine manifold provided with a vacuum, a control valve and a power unit comprising a housing having a chamber, a spring loaded piston separating said chamber into a vacuum side and an atmospheric pressure side, said control valve selectively permitting passage of atmospheric air into the pressure side of said chamber and selectively permitting a vacuum condition to be present in said atmospheric pressure side, a conduit is provided between said engine manifold and said vacuum side whereby in one selected position of said control valve a vacuum is present in said vacuum side and in said atmospheric pressure side to condition said piston to the influence of said spring and in another selected position of said control valve a vacuum is present in said vacuum side and atmospheric pressure is present in said atmospheric pressure side whereby said piston is moved in said chamber against the force of said spring, a vacuum supply tank and an emergency valve, positioned in said conduit between said engine manifold and said vacuum side whereby said engine manifold in one condition creates a vacuum in said vacuum supply tank while at the same time said engine manifold moves its said emergency valve in position to block escape of said vacuum from said vacuum supply tank and said engine manifold in another condition when atmospheric air is present in said engine manifold permits said emergency valve to move to a second position whereby said emergency valve blocks out the influence of said engine manifold on said vacuum side thereby said conduit and unblocks the escape of vacuum from said vacuum supply tank to said vacuum side to substitute the vacuum from said vacuum supply tank for the vacuum from said engine manifold.

16. An accelerator brake control consisting of an engine manifold provided with a vacuum, an accelerator, a control valve and a power unit comprising a housing having a chamber, a spring loaded piston separating said chamber into a vacuum side and an atmospheric pressure side, said control valve selectively permitting passage of atmospheric air into the pressure side of said chamber and selectively permitting a vacuum condition to be present in said atmospheric pressure side, linkage means operated by the accelerator for selectively actuating said control valve, a first means between said engine manifold and said vacuum side whereby in one selected position of said accelerator a vacuum is present in said vacuum side and through said control valve on said atmospheric pressure side to condition said piston to the influence of said spring and in another selected position of said accelerator a vacuum is present in said vacuum side and through said control valve atmospheric pressure is present in said atmospheric pressure side whereby said piston is moved in said chamber against the force of said spring, a second valve is provided, an inertia governor controlling said second valve, a second means operatively connected to said inertia governor for opening said second valve to admit atmospheric pressure to the vacuum side of said chamber.

17. In an accelerator brake control for a motor vehicle the combination of brakes, an engine manifold normally under vacuum, a power unit for applying brakes, and accelerator pedal, a first means for normally maintaining said power unit ineffective for applying the brakes, a second means connecting said manifold to said power unit for opposing said first means for maintaining the power unit ineffective, a control valve movable to two positions, in one of which it seals said power unit against atmospheric pressure, and in the other of which it admits atmospheric pressure, to overcome the said first means for maintaining the power unit ineffective in conjunction with the vacuum from said manifold, and a third means connecting said accelerator pedal to said control valve to admit atmospheric pressure during one range of its travel and to close said control valve during its movement through the driving range of said accelerator pedal, an anti-skid means for the motor vehicle is provided, comprising a fourth means for limitedly opposing the effect of said power unit on the vehicle brakes including a conduit for admitting atmospheric pressure to the vacuum side of said power unit, a valve in said conduit, inertia governor means driven by a part of said motor vehicle which rotates with the drive wheel thereof and with means cooperating therewith to maintain said valve closed when the motor vehicle wheels are in rotation, but which causes said inertia governor means to open said valve when said wheels are locked.

18. In a motor vehicle accelerator brake apparatus for hydraulic brake systems; housing means defining a pressure chamber and a vacuum chamber, atmospheric valve means selectively permitting passage of atmospheric air into the pressure chamber, equalizing valve means selectively permitting passage of atmospheric air from the pressure chamber to the vacuum chamber, linkage means operated by the accelerator for selectively actuating said valve means, whereby in one pre-determined actuation of the accelerator said atmospheric valve means is opened and said equalizing valve means is closed and whereby in a second pre-determined actuation of the accelerator said equalizing valve means is opened and said atmospheric valve means is closed and an inertia governor means in predetermined position permitting atmospheric air to enter said vacuum chamber when said equalizing valve means is closed.

19. In a motor vehicle accelerator brake apparatus for hydraulic brake systems; housing means defining a pressure chamber and a vacuum chamber, atmospheric valve means selectively permitting passage of atmospheric air into the pressure chamber, equalizing valve means selectively permitting passage of atmospheric air from the pressure chamber to the vacuum chamber, linkage means operated by the accelerator for selectively actuating said valve means, whereby in one pre-determined actuation of the accelerator said atmospheric valve means is opened and said equalizing valve means is closed and whereby in a second pre-determined actuation of the accelerator said equalizing valve means is opened and said atmospheric valve means is closed, an engine manifold, a vacuum supply tank, an emergency valve, said engine manifold supplying vacuum to said vacuum chamber, said emergency valve actuating, upon failure of said engine manifold to supply vacuum to said vacuum chamber, to supply vacuum from said vacuum supply tank to said vacuum chamber.

20. An accelerator brake control consisting of an engine manifold normally under vacuum, a power unit, an accelerator, a first means for normally maintaining said power unit ineffective, a second means connecting said manifold with said power unit for opposing said first means for maintaining said power unit ineffective, a control valve movable to two positions, in one of which it seals said power unit against atmospheric pressure and in another position of which it admits said atmospheric pressure to overcome said first means for maintaining said power unit ineffective in conjunction with the vacuum from said engine manifold, a third means connecting said accelerator to said control valve to admit atmospheric pressure during one range of the travel of said accelerator and to close said control valve to atmospheric pressure during another range of travel of said

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accelerator, a vacuum supply tank, an emergency valve, said vacuum supply tank replacing said vacuum normally supplied by said engine manifold upon failure of said engine manifold to supply a vacuum, said failure actuating said emergency valve to unite said second means with said vacuum supply tank and to disconnect said second means with said engine manifold.

21. In a power brake system, a power unit, a set of brakes, a closed circuit fluid supply between said power unit and set of brakes, an emergency valve, a solenoid check valve, an electrical circuit comprising a source of electrical energy, a first switch, said solenoid in the solenoid check valve and a second switch actuated by said emergency valve, a source of fluid power to activate and deactivate said power unit to actuate and de-activate said set of brakes through said closed circuit fluid supply, said electrical circuit normally energized, failure of said source of power actuating said emergency valve to close said second switch to complete said electrical circuit to energize said solenoid to actuate said solenoid check valve to permit fluid in said closed circuit to flow toward said set of brakes, said solenoid check valve preventing fluid in said closed circuit from flowing from said set of brakes to said power unit until said first switch de-energized said electrical circuit.

22. An accelerator brake control consisting of an engine manifold provided with a vacuum, a control valve, an emergency valve, a check valve, a solenoid check valve, and a power unit comprising a housing, a hydraulic fluid cylinder former in said housing, said housing having a chamber, a first piston separating said chamber into a vacuum side and an atmospheric pressure side, said first piston extending into said hydraulic fluid chamber, a fluid conduit connecting said solenoid check valve with said hydraulic fluid cylinder, said control valve selectively permitting passage of atmospheric air into the pressure side of said chamber and selectively permitting a vacuum condition to be present in said atmospheric pressure side, a conduit provided with said check valve connecting said engine manifold with said vacuum side, said emergency valve comprising a chamber, a diaphragm dividing said chamber into a vacuum side and an atmospheric pressure side, a second piston secured to said diaphragm slidably mounted in said atmospheric pressure side, a spring urging said second piston and diaphragm toward said pressure side, a plate of electrical conductivity and electrically insulated from said second piston, secured to said second piston, electrical terminals fixed to said emergency valve, engageable and disengageable with said plate, an electrical circuit having a source of electrical power, comprising a switch, the solenoid of said solenoid valve and said terminals, a second conduit connecting said engine manifold with said vacuum side of said emergency valve, a third conduit leading from said solenoid check valve, said control valve in one selected position permitting vacuum from said engine manifold to be present in said vacuum side of said power unit and atmospheric pressure in said atmospheric pressure side of said power unit to move said first piston to force hydraulic fluid from said hydraulic fluid chamber through said solenoid check valve into said third conduit, said control valve in another selected position permitting atmospheric pressure to be present in both said vacuum side and atmospheric pressure side of said power unit, a spring in said vacuum side urging said first piston toward said atmospheric pressure side, said check valve operatively preventing atmospheric pressure in said vacuum side from reaching said manifold, upon vacuum failure in said engine manifold with said electrical circuit energized, said spring will urge said diaphragm and said second piston forward to engage said plate with said terminals to cause said solenoid check valve to block fluid from said third conduit from entering said solenoid check valve but allowing fluid from said hydraulic fluid cylinder to pass through said fluid conduit to said solenoid check valve into said third conduit, upon de-energizing said electrical circuit, said solenoid check valve remains in position to block passage of fluid from said third conduit until said first piston is actuated to force fluid from said hydraulic fluid chamber into said solenoid check valve with the solenoid de-energized.

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BRAKING SYSTEM AND ANTI-SKID MEANS THEREFOR

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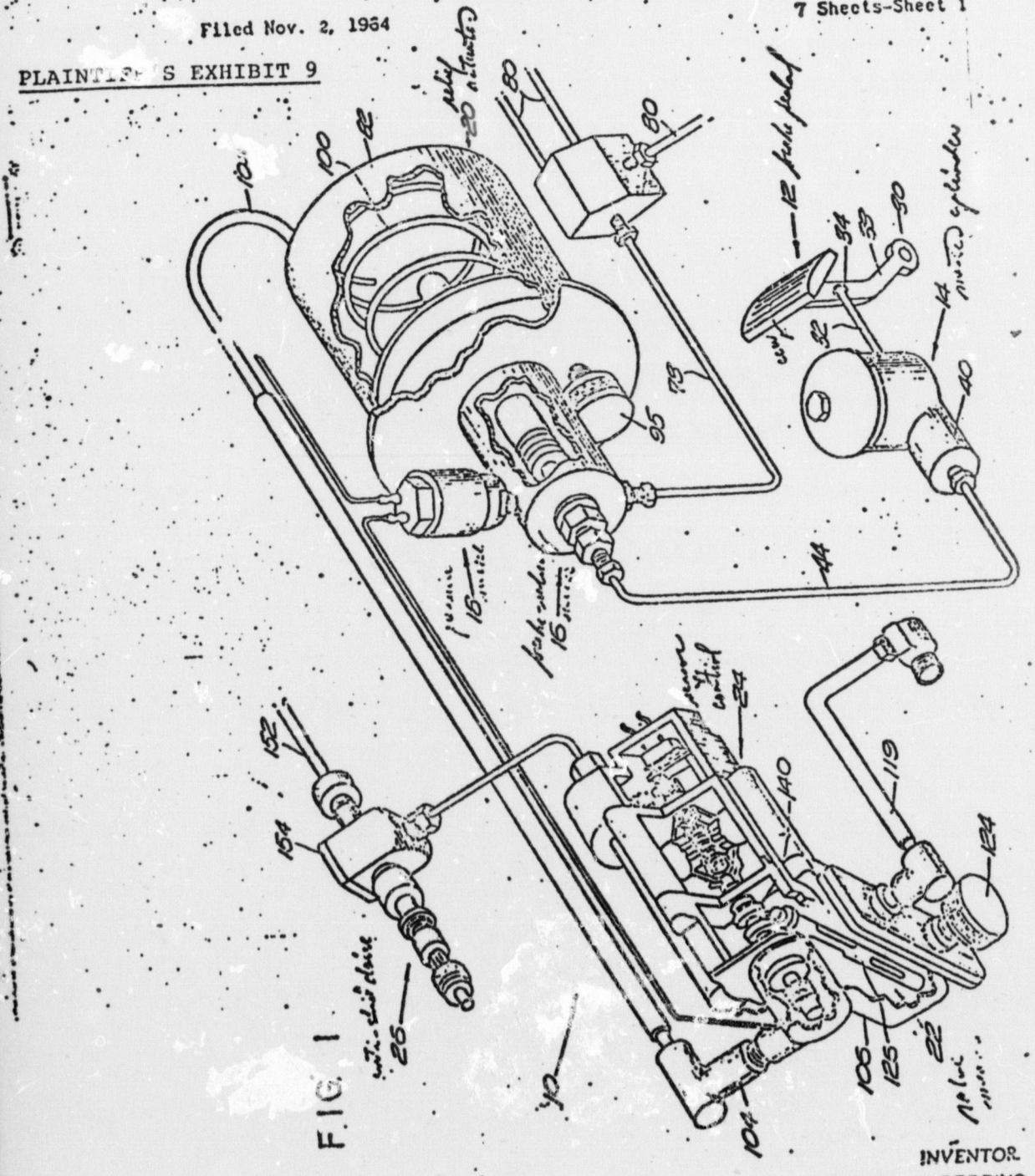


FIG. 1

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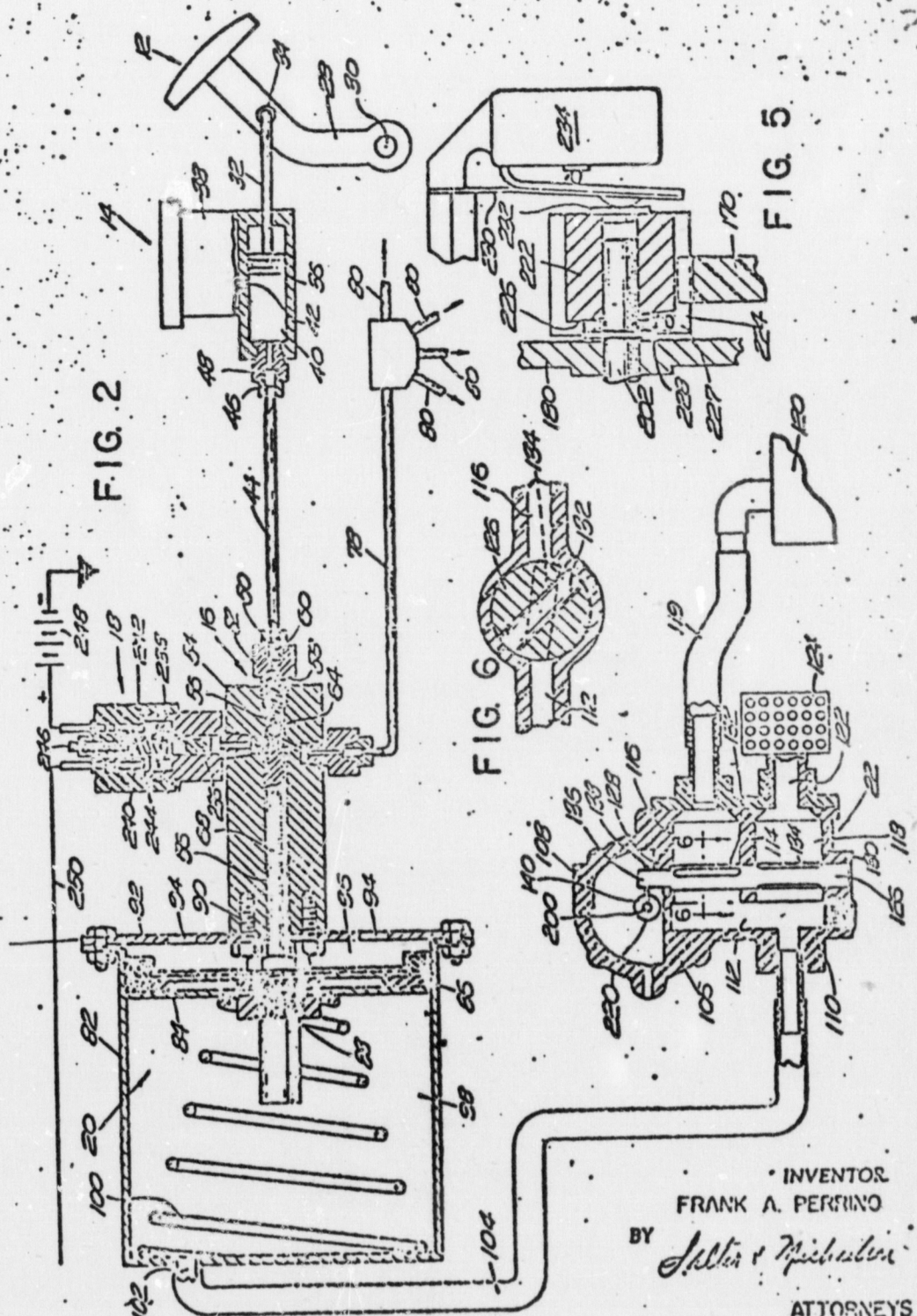


FIG. 2

FIG. 5

FIG. 6

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BRAKING SYSTEM AND ANTI-SKID MEANS THEREFOR

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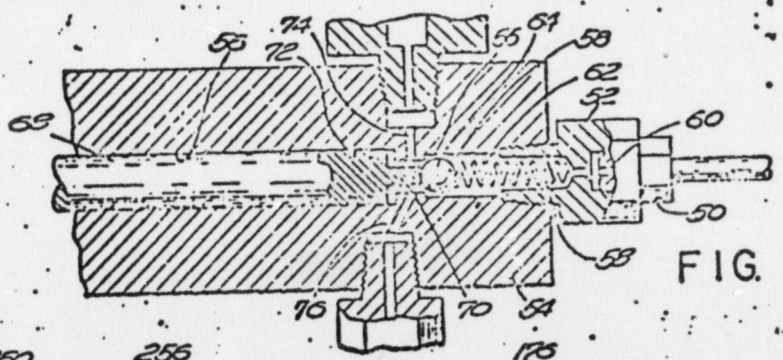


FIG. 3

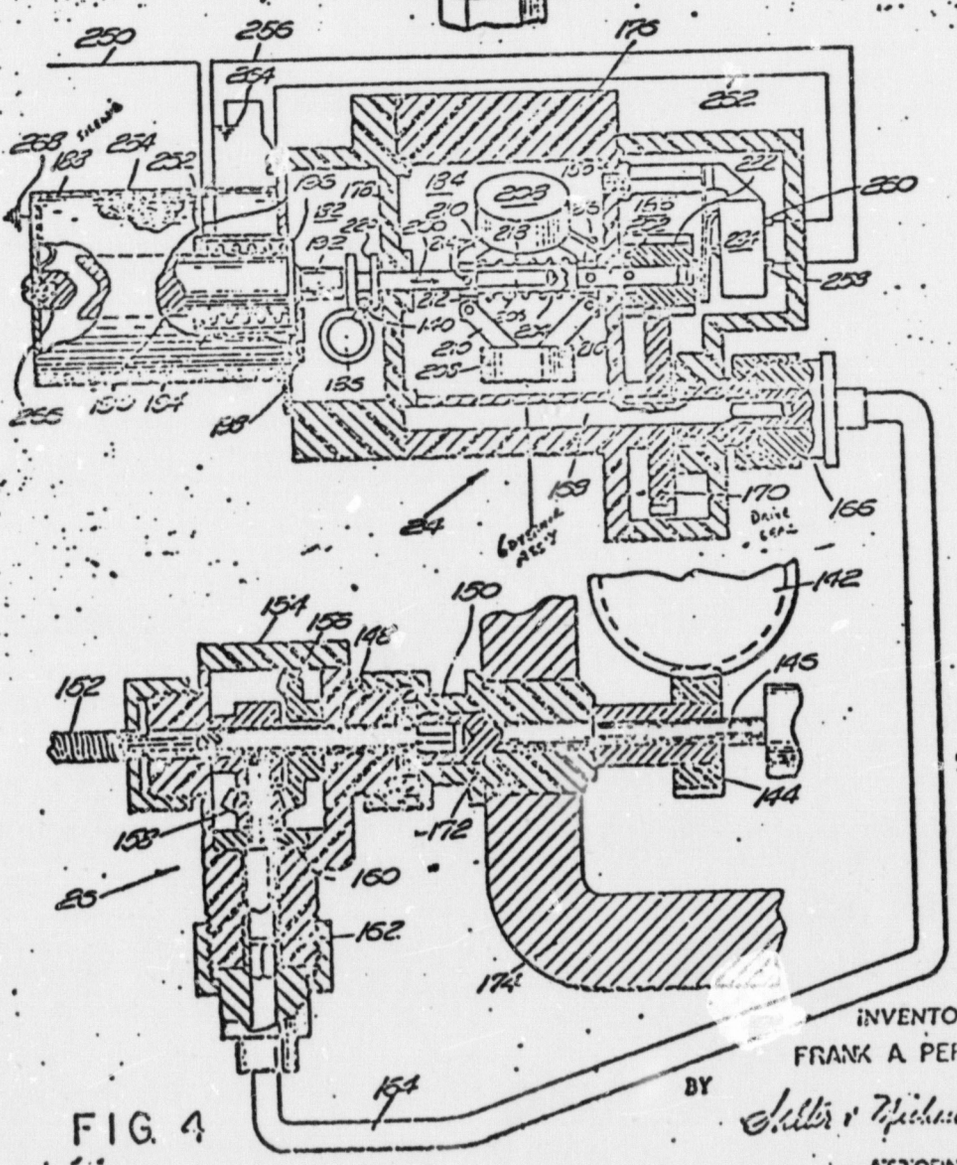


FIG. 4

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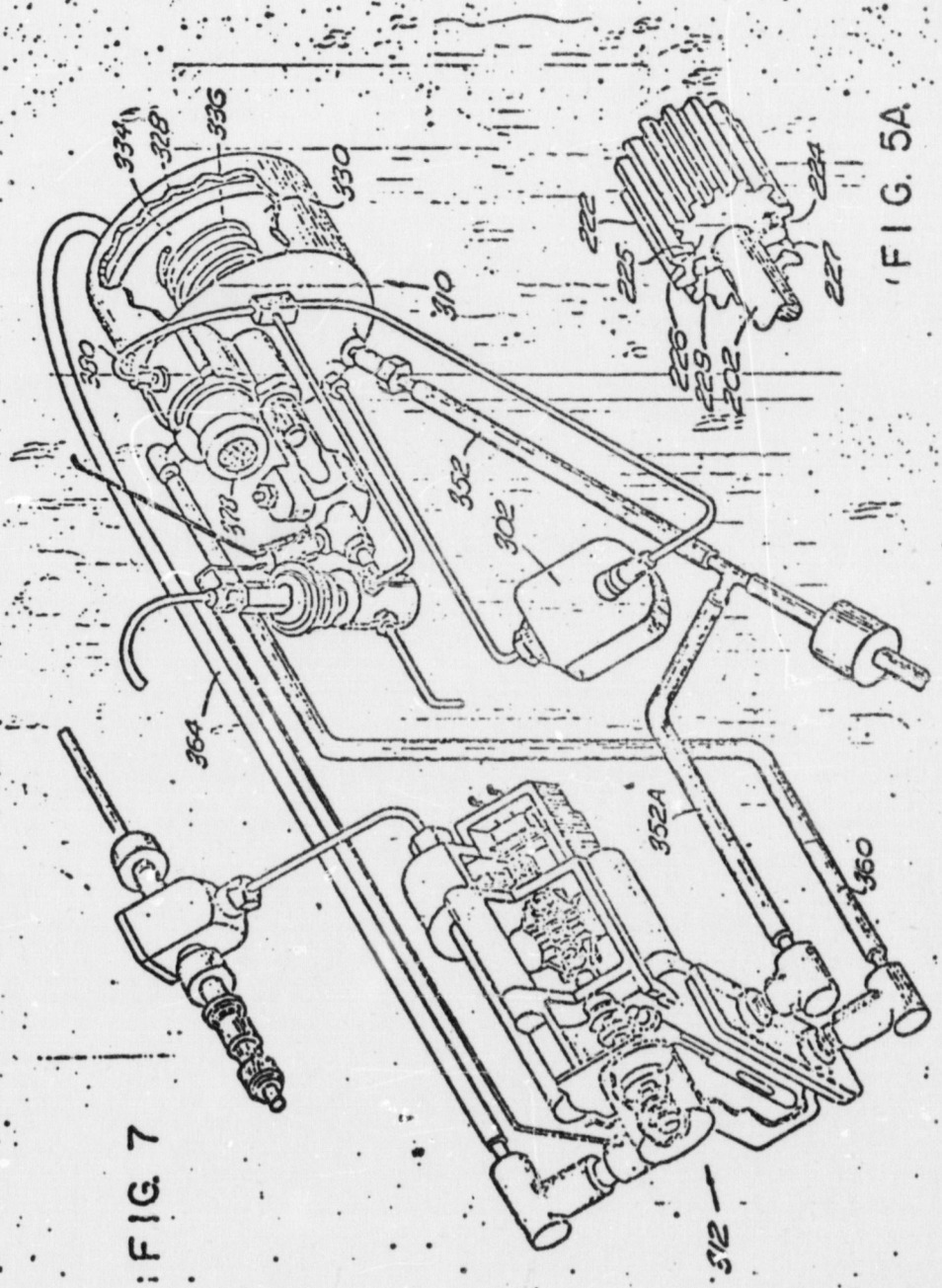


FIG. 5A

FIG. 7

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BRAKING SYSTEM AND ANTI-SKID MEANS THEREFOR

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FIG. 3

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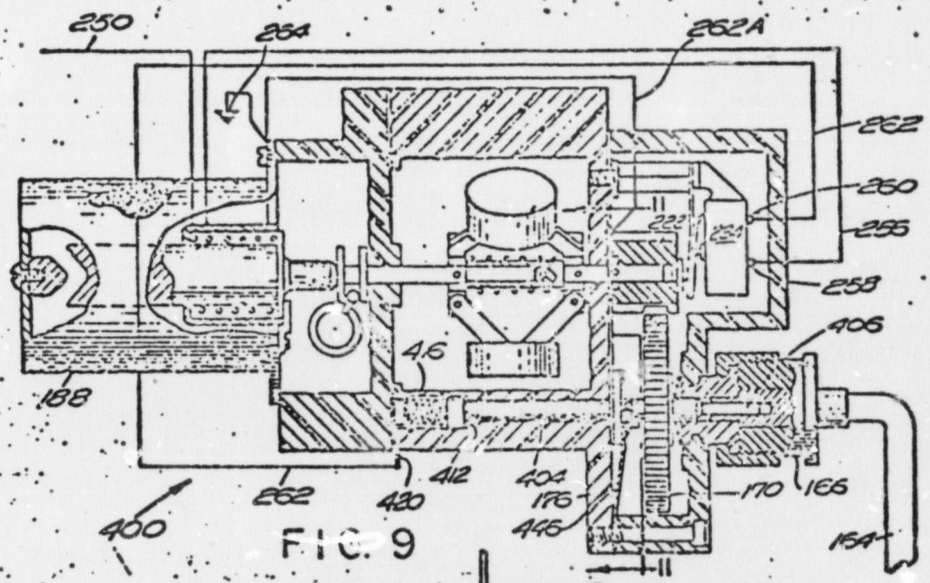


FIG. 9

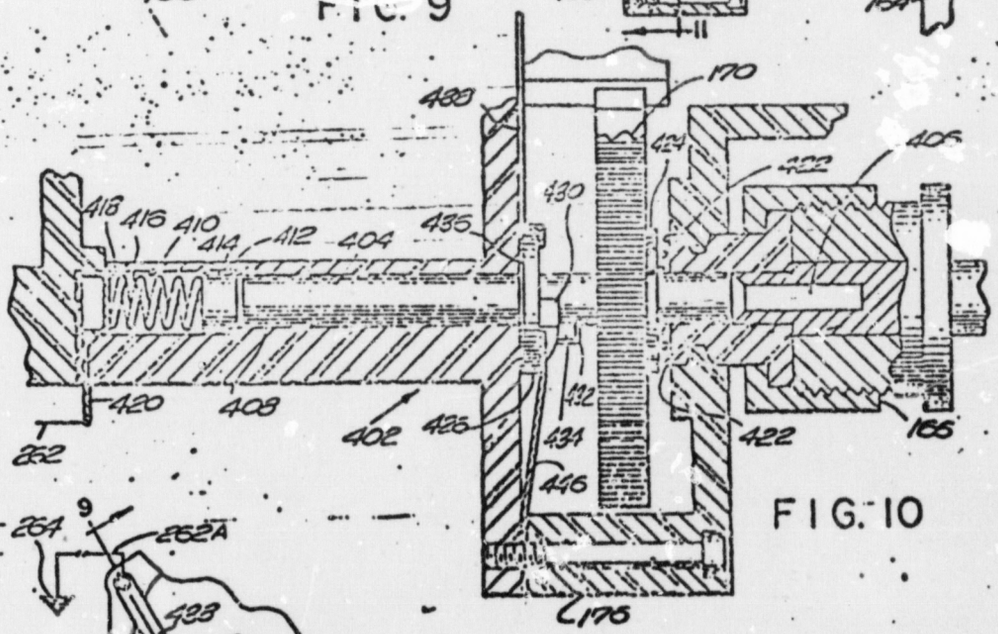


FIG. 10

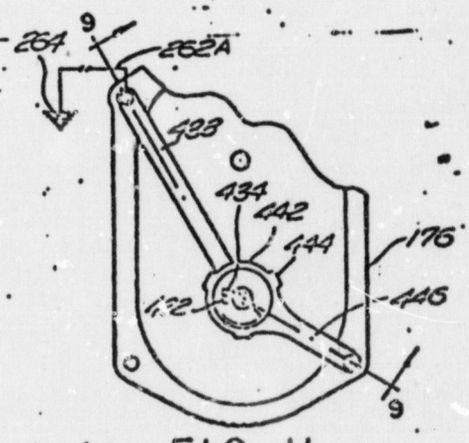


FIG. 11

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FIG. 12

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BRAKING SYSTEM AND ANTI-SKID MEANS THEREFOR

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Filed Nov. 2, 1964, Ser. No. 409,350
23 Claims. (Cl. 303—21)

This is a continuation-in-part of applicant's copending application Ser. No. 322,716, filed Nov. 12, 1963, and entitled, Braking System and Anti-Skid Means Therefor.

The present invention relates generally to braking systems for automotive vehicles and the like and is particularly concerned with the provision of novel and improved anti-skid means.

It is well known that a skid condition may be encountered by a moving vehicle where the brakes of the vehicle are applied so hard that the vehicle wheels tend to lock. Quite obviously, other factors affect the commencement and existence of a skid condition, such as the condition of the surface on which the vehicle is moving, the speed at which the vehicle is traveling, etc. However, it is possible for a vehicle to commence a skid even where the surface on which the vehicle is traveling is not wet or icy and even where the vehicle is traveling at moderate speeds, if the vehicle brakes are applied too hard. Since all control of a vehicle is lost while the vehicle is in a skid, thus increasing the danger of serious accident and personal injury, it is obviously highly desirable to prevent or reduce skidding while at the same time enabling the vehicle to come to a complete stop as quickly as possible.

It is therefore a primary object of the present invention to provide an improved braking system wherein operation of the brakes is under control of the vehicle operator until a skid condition commences to exist, at which time anti-skid means automatically take over complete control of the brakes and maintain said control until danger of skidding no longer exists, whereupon the brakes are returned to normal control by the operator.

Another object of my invention is the provision of efficient and sensitive control means for the various operations of the system, which have usefulness in other types of systems as well.

Another object of my invention is the provision of novel and improved anti-skid means equally adaptable to so-called power brakes, as well as nonpower brakes.

A further object of my invention is the provision of anti-skid means of the character described wherein the vacuum system of the vehicle is utilized, but wherein the loss of vacuum in no way interferes with control of the vehicle brakes by the operator.

A further object of my invention is the provision of anti-skid means that is readily and easily adaptable to existing vehicles.

Another object is the provision of a novel and improved reverse cut-off switch that automatically renders the anti-skid means inoperative when the vehicle moves in reverse.

Still another object of my invention is the provision of an anti-skid system for automotive vehicles and the like having means which are durable and effective in operation and which are feasible and practical both from an economic and manufacturing standpoint.

Other objects, features and advantages of my invention will become apparent as the description thereof proceeds when considered in connection with the accompanying illustrative drawings.

In the drawings, which illustrate the best mode presently contemplated for carrying out the present invention:

FIG. 1 is a diagrammatic view, in perspective, of the assembly utilized in my nonpower unit, with portions being broken away for purposes of illustration;

FIG. 2 is an elevational view, in section, of a portion of the overall assembly illustrated in FIG. 1;

FIG. 3 is an enlarged fragmentary sectional elevation of a portion of the brake-actuating means illustrated in FIG. 1;

FIG. 4 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid unit and the drive means therefor;

FIG. 5 is an enlarged sectional elevation of the cam gear and solenoid-actuating switch which form a part of the anti-skid unit;

FIG. 5A is an enlarged fragmentary perspective view of the cam gear;

FIG. 6 is an enlarged section taken on line 6—6 of FIG. 2;

FIG. 7 is a diagrammatic view, in perspective, of my anti-skid means in combination with a power brake unit, portions being broken away for purposes of illustration;

FIG. 8 is an elevational view, in section, of the power brake assembly illustrated in FIG. 7;

FIG. 9 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid means and the reverse cut-off switch in combination therewith;

FIG. 10 is an enlarged fragmentary sectional elevation of the reverse cut-off switch per se;

FIG. 11 is a fragmentary section taken on line 11—11 of FIG. 9; and

FIG. 12 is an enlarged exploded perspective view of the reverse cut-off switch per se.

According to one aspect of my invention, a sensing means for sensing when a skid condition begins is caused to activate a wheel speed-sensitive device that takes over control of the brake pressure until the skid condition ends. Advantageously, the sensing means is an inertia-responsive means that senses a sudden change in the speed of rotation of a wheel at the commencement of skid conditions. This inertia-responsive device does not control the brake pressure itself by repeated action during the skid condition, as has been attempted by the prior art. Instead the control of the brake pressure is taken over completely by a governor driven dependently with the wheel, advantageously a mechanical governor. The governor then controls the brake pressure in a hunting or modulating manner, for an extended period, until such control of the brakes is ended by a reset means that responds to slowing of the vehicle to a predetermined speed. Preferably, the reset means is a brake-pressure-sensitive device which senses reduction of brake pressure to a predetermined level which the governor can produce only when it slows to the predetermined speed.

This separation of the initiating and controlling functions is an extremely important feature. The initiating device must necessarily have a restraint to prevent operation when the vehicle merely decelerates normally I have realized that such a restraint prevents the device from being able to effectively regulate the brake pressure due to lack of sensitivity. The governor, after it takes over control, is free of such restraint and therefore can have much greater sensitivity with the result that the brake pressure can be much more accurately regulated.

The reset device is also of critical importance because at low speeds, e.g., below 10 m.p.h., the governor forces can become so diminished that the governor can behave entirely as if the vehicle were skidding, though not, thereby totally relieving the brake pressure. The reset device can act before this occurs, whereby normal control is regained.

A further aspect of my invention concerns the realization that a predetermined level of brake pressure, e.g., the minimum brake pressure at which skidding of the vehicle will occur on ice, generally between 100-150 p.s.i.

for conventional automobiles, can be utilized to determine the setting for the brake pressure-sensitive reset device. Thereby, only at the time when skidding is no longer possible will control be taken away from the anti-skid unit to return the brake system to normal operation. This pressure-sensitive device, particularly when taking the form of a brake fluid pressure-sensitive switch, is useful with anti-skid systems generally, though of particular usefulness in the system described above.

Advantageously, the brake pressure-sensitive switch operates an electrical system, preferably a solenoid, to restore normal brake operation, and it can be employed to bring the anti-skid unit into readiness whenever the operator applies his brakes, as well as to stop the anti-skid unit from acting when skid conditions have ended.

Another aspect of the invention concerns the control of the governor. In normal operation of the brakes, the governor is prevented from influencing the brakes, but once activated, e.g., by the inertia-sensitive device, it is made continually to act, even when the vehicle wheel speeds up after the brakes have been initially relieved. A controlling or restraining member, preferably a solenoid, is employed to achieve this operation, with the controlling member held steadily from interfering with the governor until the reset device operates. In the case where a rotary weight mechanical governor is employed, a plunger can normally restrain the governor and a solenoid, initially under the control of the inertia-responsive device, can pull back the plunger to free the governor, whereupon the plunger can operate a switch to maintain the solenoid energized, and at the end of skid conditions the reset device can de-energize the solenoid to once again restrain the governor.

Another aspect of the invention is a fluid valving system that responds to the governor to increase and decrease brake pressure. Advantageously, the valve, discharging to a brake control conduit, is moved by the governor alternately between two positions to connect two passages for fluid at different pressures to the conduit. This causes the brake pressure to rapidly vary in a modulating or hunting manner, to keep the brake pressure at maximum values while preventing skidding. The means about which the pressure modulates changes as the slipperiness of the road changes, due to the effect such road changes have upon the governor.

For use in a closed-circuit hydraulic system, the direct control of the brakes is achieved, according to the invention, by an isolation valve that continually seals the wheel brake from the control of the operator, and a variable volume chamber, e.g., piston and cylinder, for relieving and reapplying brake pressure to the wheel, the piston modulating about a mean determined by the modulating pressure conditions of the brake control conduit. Advantageously, the piston, at the position where it allows full pressure, can operate the isolation valve. Then the piston being spring loaded to that position also can perform a pressure-limiting function, limiting brake pressure to, e.g., 1,000 p.s.i., though the operator is capable of applying twice that much. In normal operation the brakes function as well at 1,000 p.s.i. as above, and by so limiting the brake pressure, less pressure is present to be relieved when a skid occurs, which enables faster action and allows the relief chamber to be smaller.

In other brake systems the brake control conduit can control a power unit, or in the case of air brakes, it can directly control the brake actuators associated with the wheels.

Furthermore, according to the invention, I have provided advantageous and improved constructions of the various operative components and sub-systems which can best be described with reference to the drawings.

Referring now to the drawings, and more particularly to FIGS. 1-6 thereof, a closed hydraulic circuit braking system employing my novel and improved anti-skid means in combination with a so-called nonpower brake

will be described. The over-all system, generally designated as 10, comprises brake pedal 12, master cylinder 14, brake-relieving device 16, hydraulic pressure switch 18, relief actuator means 20, conduit valve means 22, skid-sensing and control means 24, and anti-skid unit drive means 26.

Referring to FIGS. 1 and 2, the brake pedal 12 is of conventional construction and comprises bell crank 28 pivotally connected to a suitable support on the vehicle (not shown) at 30 whereupon depression of pedal 12 causes bell crank 28 to rotate in a counterclockwise direction around pivot point 30, thus actuating connecting rod 32 pivotally connected to bell crank 28, as at 34. Rod 32 carries at its other extremity a piston 36 which functions to pump oil or other suitable fluid from master cylinder 14, all in a well-known manned. Specifically, master cylinder 14 has an upper reservoir 38 which communicates with lower chamber 40 by means of opening 42 whereupon a full supply of oil is always present in the chamber 40 to be pumped by actuation of pedal 12 and piston 36.

Fluid pumped from master cylinder 14 passes through a conduit 44 connected at one of its extremities by suitable couplings 46, 48 to chamber 40 of master cylinder 14 and at its other extremity by a suitable coupling 50 to the brake-relieving device generally designated at 16, which incorporates an isolation valve. More specifically, coupling 50 is connected to a second coupling 52 which is threadedly attached to block 54, said block having a bore 56 extending longitudinally therethrough into communication with aligned longitudinally extending bores 58 and 60 in couplings 52 and 50, respectively.

As will be seen most clearly in FIGS. 2 and 3, coupling 52 is provided with an internal shoulder for receiving positioning spring 62 which positions isolation valve ball 64 adjacent valve seat 66, tending to close with slight force the bore or passageway 56. During normal operation of the vehicle, ball 64 is maintained in its unseated position, as illustrated in FIGS. 2 and 3, against the action of spring 62, by means of spring-loaded actuator piston 68 having a reduced extension 70 in engagement with ball 64. Suitable O-rings or the like 72 are provided to insure a tight sliding fit between piston 68 and bore 56 whereby said piston effectively blocks further flow of fluid through bore 56.

Block 54 is provided with a pair of oppositely disposed passageways 74 and 76, it being noted that said passageways communicate with bore 56 just behind valve seat 66, whereupon when ball 64 is in its closed position in engagement with valve seat 66, fluid in conduit 44 from the master cylinder will be blocked before reaching the passageways 74 and 76. Passageway 74 communicates with hydraulic pressure reset switch 18 for reasons hereinafter to be described, while passageway 76 communicates by means of suitable couplings with conduit 78, which in turn communicates with the hydraulic brake lines 80 leading to the wheel hydraulic actuator cylinders (not shown) of the vehicle to actuate the brakes in well-known fashion. Thus it will be seen that with the parts in the positions illustrated in FIGS. 2 and 3, and particularly with piston 68 in the position illustrated, brake pedal 12 controls actuation of the vehicle brakes through a direct line. This is the normal operating condition of the vehicle.

The actuator for the relieving device 16, generally designated at 20 (FIGS. 1, 2) controls the movement of piston 68 and hence ball 64. The actuator comprises a housing or cylinder 82 having a large piston 84 slidably mounted therein, said piston having suitable seals 86 at its outer periphery to insure that the piston 84 will make a tight sliding fit within cylinder 82 whereby no leakage will exist around the piston. Piston 84 is threadedly connected, as at 88, to the aforementioned small hydraulic piston 68, whereby movement of large piston 84 causes corresponding movement of piston 68.

Housing 82 is secured to block 54 by any suitable means, such as screws 90, and the adjacent end wall 92 of the housing 82 is in free communication with atmosphere, as by ports 94. Suitable air filters, such as shown at 95 in FIG. 1, may be associated with the ports 94. It will thus be seen that the chamber 96 located on one side of large piston 84 is always at atmospheric pressure, while chamber 98, located on the opposite side of piston 84, may either be at atmospheric pressure or it may be vacuum charged, as will hereinafter be described in more detail. When the chambers 96 and 98 are both at atmospheric pressure, spring 100 resiliently urges piston 84 to the position illustrated in FIG. 2, it being seen that the piston 84 has moved to the right as far as it can go. In this position, the piston 68 has also been axially moved to the position illustrated, in which position extension 70 (FIG. 3) has engaged ball 64 to unseat it from valve seat 66. Thus, where chambers 96 and 98 are both at atmospheric pressure, the vehicle brakes are under the complete control of the brake pedal 12. However, when chamber 98 is vacuum charged, the pressure differential between chambers 96 and 98 is sufficient to overcome spring 100, whereupon piston 84 will move to the left, viewing FIG. 2, thus causing corresponding movement to the left of piston 68. This in turn enables ball 64 to seat against valve seat 66, by means of spring 62, thus blocking the bore 56 and preventing actuation of the brakes by means of brake pedal 12. As pistons 84 and 68 move further to the left due to the presence of a vacuum or even a partial vacuum in chamber 98, hydraulic pressure in lines 78 and 80 and in the spring-loaded brake actuators associated with the vehicle wheels will be relieved, due to the fact that the fluid is now free to back up into that portion of bore 56 that has been vacated by movement of piston 68 to the left. It will therefore be seen that by introducing a vacuum or partial vacuum to chamber 98, fluid pressure in lines 78 and 80 will automatically be relieved, thereby relieving the brakes. By the same token, as soon as atmosphere is again introduced to chamber 98 to equalize the pressure in chamber 96, the brakes can once again be fully actuated by the pedal 12. It will be apparent that the amount of vacuum introduced to chamber 98 will determine the amount of reduction in brake pressure. More specifically, the greater the vacuum that is introduced to chamber 98, the greater will be the movement of piston 84 to the left (viewing FIG. 2) and at the same time there will be greater movement of piston 68 to the left, thus vacating more of bore 56 and providing greater relief of the hydraulic brake pressure. Similarly, given a certain amount of relief due to a certain amount of vacuum, then decrease in the amount of vacuum will increase the hydraulic brake pressure due to movement of the pistons to the right. The ability of my system to provide varying degrees of relief and restoration of brake pressure while the brake pedal remains isolated from the system is an important and advantageous feature.

It will be noted that during normal operation of the system, as heretofore described, i.e., where atmospheric pressure exists in both chambers 96 and 98, the piston 68 will function as a pressure limitor in the system. More specifically, since the piston 68 is resiliently held in the position illustrated in FIG. 2 by means of spring 100, it will back off when excess pressure is applied by foot pedal 12. In practice it has been found that in a conventional automobile pressure above 1,000 p.s.i. or at least above 1,500 p.s.i., depending on the type of vehicle, is unneeded for normal brake operation and hence springs 62 and 100 are preferably calibrated so as to enable ball 64 to seat when pressure in excess of that needed is introduced, thus preventing introduction of excessive pressure to the brakes. It will be obvious that the springs may be calibrated so as to provide a limit at any selected pressure.

The means for introducing either atmospheric pressure or vacuum to chamber 98 and for modulating the degree of vacuum therein will now be described. Referring to FIG. 2, it will be seen that cylinder housing 82 is provided with an opening 102 in communication with chamber 98, which opening communicates with a brake control conduit 104 that in turn connects with the conduit valve, generally designated at 22. More specifically, there is provided a housing 106 having an upper portion 108 which houses the skid-sensing and control means 24, and a lower portion 110 which houses the valve 22. The lower housing 110 comprises an inner chamber 112 having a partition 114 defining a vacuum passage 116 and an atmosphere passage 118. More specifically, vacuum passage 116 is in communication, by means of passage 119, with the intake manifold 120 of the vehicle. Thus, suction is constantly being applied to the passage 116. The passage 118, on the other hand, is in communication with atmosphere through port 122, to which a suitable air filter 124 may be attached, if desired. A rotary valve 126 is mounted for rotation in housing 106, it being noted that valve 126 is an elongated rod that extends vertically through upper partition 128, the aforesaid partition 114, and is journaled in bottom wall 130, the partition and wall being integral parts of a unique molded housing of antifriction plastic. The valve 126 is provided with a pair of passageways or bores 132 and 134 extending diametrically therethrough, said bores being vertically spaced from each other, and having elongated cross-sections in the axial direction, longer than the diameter of the rod. Bore 132 is associated with passage 116 and bore 134 with passage 118. The bores are angularly disposed with respect to each other, as shown most clearly in FIG. 6. Thus, with the rotary valve 126 in the position illustrated in FIGS. 2 and 6, bore 134 is positioned so as to allow free passage of atmosphere from passages 122 and 118, bore 134 and chamber 112 to brake control conduit 104, whereby atmosphere will flow into chamber 98. At the same time, due to the angular displacement of bore 132, passage 116 will be closed by valve 126, thus preventing suction from manifold 120 and passage 119 from coming into contact with chamber 112 and brake control conduit 104. It will be understood that the only access from passages 116 and 118 to chamber 112 is through valve 126 and more specifically, the bores 132, 134 thereof. Should valve 126 be rotated by means hereinafter to be described, it will be seen that communication between passage 118 and chamber 112 will eventually be blocked, thus interrupting the flow of atmosphere to brake control conduit 104, while at the same time passage 116 will be in communication with chamber 112, whereupon a suction will be applied through brake control conduit 104 to chamber 98.

It is important to note that use of a rotary valve such as valve 126 is extremely advantageous where passage of a vacuum and atmospheric pressure, or more generally two different pressures, is being controlled due to the fact that the pressure differential offers a minimum of resistance to the turning movement of the small diameter valve. This is contrasted to a reciprocal or poppet type valve wherein movement of the valve is much more resisted by pressure differential.

As hereinbefore explained, and described, introduction of a vacuum or partial vacuum to chamber 98 automatically tends to release brake pressure on the vehicle wheels. It has been found in practice that an angular displacement between bores 132 and 134 of approximately 37.5° gives best results in the operation of my system, and it will be understood that rotary valve 126 may not always be positioned so as to completely open or close the bores 132 and 134. Thus, different degrees of movement of valve 126 allow different brake fluid pressures to be exerted on the vehicle wheels, thus providing the proper braking pressures for different road conditions.

In instances where the pressure on the brake actuator is the same at the beginning of a skid (lesser initial pressure requires less relieving), the amount of relief of the brake pressure depends on the amounts of travel of piston 84 to the left in chamber 98 which in turn is dependent on the amount of vacuum introduced to the chamber. Thus, where the vehicle is braked and skids on a dry surface which requires a greater brake pressure to create a skid condition, lesser movement of piston 84 to the left, and hence less vacuum, will be required to relieve the brake pressure sufficiently to prevent the vehicle wheels from locking, this pressure being referred to as "release" pressure. Conversely, where the vehicle is braked on a slippery surface such as ice, a lesser brake pressure is required to create a skid condition. Hence the valve 126 must open suction passages 116 to the brake control conduit 104 for a longer initial period to provide greater vacuum to chamber 98 in order to move piston 84 sufficiently to the left so that sufficient relief of the brake pressure will be effected to prevent the wheels from locking.

In accordance with the invention after release pressure is attained, with corresponding movement of the piston then the governor causes the degree of vacuum in conduit 104 to modulate, which modulates the brake pressure through a narrow range of intermediate pressures above and below the release pressure, without ever restoring the normal, higher brake pressure until the skid condition ends.

To thus operate the valve, the upper extremity of valve 126 (FIG. 2) is provided with a reduced portion 136 that extends upwardly through partition 128, said portion 136 having a pin 138 extending therethrough to which is attached a circular projection 140, it being seen that as a longitudinal thrust is exerted against projection 140, the valve 126 will be caused to rotate.

The means for controlling movement of valve 126, namely, the skid-sensing and governor control means and the drive means therefor, will now be described. Referring to FIG. 4, skid-sensing and governor control means is designated generally at 24, and the drive means therefor is designated generally at 26. Referring first to the latter, the vehicle drive shaft is shown at 142, it being understood that the drive shaft is directly connected to the wheels of the vehicle (not shown) whereupon when the vehicle wheels lock, the drive shaft also locks. The drive shaft drivingly engages speedometer gear 144, which in turn is fixed to shaft 146, which in turn is drivingly connected to shaft 148 by means of a coupling shown at 150, said shaft 148 connecting with speedometer cable 152. Gear box 154 through which shaft 148 passes and in which the shaft 148 is suitably journaled, houses drivingly engaged bevel gears 156, 158, it being understood that bevel gear 156 is keyed to shaft 148 and bevel gear 158 is keyed to shaft 160, whereupon rotation of shaft 148, transmitted thereto by the vehicle drive shaft 142, is transmitted to shaft 160. Shaft 160, in turn, is drivingly interconnected, as at 162, to flexible shaft 164, said flexible shaft being connected, as at 166, to the unit 24, and said flexible shaft being drivingly interconnected with a shaft 168 which has fixed thereto a drive gear 170. It will therefore be seen that the gear 170 is drivingly interconnected with the vehicle drive shaft 142, which in turn is connected to the vehicle wheels. Thus, rotation of the vehicle wheels will cause corresponding rotation of gear 170, and, conversely, sudden slowing of the vehicle wheels, such as would occur where the vehicle approaches a skid condition by the operator applying more brake pressure than is required for a particular road surface, will cause not only the main drive shaft 142 to suddenly slow, but also drive gear 170. Gear box 154, and adaptor 172, by means of which the gear box is connected to the engine housing 174, comprise an assembly which may be easily and readily adapted to a conventional vehicle for driving the sensing and governor control means now to be described.

The unit 24 (FIGS. 1, 4) comprises a molded plastic housing 176 having a pair of spaced partitions 178, 180 therein defining chambers 182, 184 and 186. A solenoid 188 is secured to housing 176 adjacent chamber 182 by any suitable means, the solenoid 188 having a plunger 190 with a reduced extension 192 resiliently urged into chamber 182 by means of spring 194. When solenoid 188 is not energized, spring 194 bears against flange 196 affixed to plunger 190 to resiliently urge extension 192 into chamber 182. Suitable stop means 198 mounted on the interior of chamber 182 engaged flange 196 to limit resilient axial movement of plunger 190 and extension 192 at a predetermined point, for reasons hereinafter to be explained more fully. Journaled in the partitions 178 and 180 are a pair of aligned shafts 200 and 202, respectively. Shaft 202 is provided with an enlarged portion 204 having an internal bore for slidably receiving in telescoping relation a reduced extension 206 of shaft 200. Attached to shafts 200 and 202 is a governor assembly comprising a plurality of weights 208, each pivotally connected to a pair of links 210, which links in turn are pivotally connected to arms 212. The arms 212 located on one side of the weights 208 are affixed to shaft 200 as at 214, while the arms on the other side of the weights are affixed to shaft 202 as at 216. As hereinbefore pointed out, shaft 202 is journaled for rotation in partition 180 but is incapable of longitudinal movement. Shaft 200, on the other hand, is drivingly engaged with shaft 202 so as to rotate therewith, but said shaft 200 is axially slidable with respect to the said shaft 202. Since the governor assembly is affixed on one side to shaft 202 and on the other side to shaft 200, it will be seen that the speed of rotation of the shafts will determine their longitudinal position relative to each other. Spring 218 resiliently urges the governor assembly to its collapsed position wherein shafts 200 and 202 are axially extended with respect to each other, but as the rotational speed of shafts 200 and 202 increases, centrifugal force urges the weights 208 outwardly, thus moving shaft 200 toward the right, viewing FIG. 4, against the action of spring 218.

Fixed to shaft 200 at the extremity thereof located in chamber 182 is a spool member 220, which spool member is in engagement with projection 140 carried by the reduced upper extension 136 of rotary valve 126. Thus, axial movement of shaft 200 causes like movement of the spool 220 which in turn carries therewith projection 140 captured between the opposed walls of the spool to thereby impart rotary movement to valve 126.

The means for imparting rotational movement to shafts 200 and 202 and the governor assembly associated therewith comprises a cam gear 222 (FIGS. 4, 5 and 5A), which cam gear is part of the drive train, in driven engagement with gear 170. The cam gear 222 is mounted on shaft 202 and is located in chamber 186. The cam gear is slidably mounted on shaft 202 and is in driving engagement therewith by means now to be described. A recess 224 having inclined or helical surface 226 is provided at the end of the cam gear adjacent partition 180, and a projection or pin 228, carried by shaft 202, extends into said recess in engagement with the cam surface 226. At either end of the recess is a stop surface, at the deeper end stop surface 227, adapted to engage pin 228 for accelerating the governor and at the shallower end stop surface 225 for engaging pin 228 to decelerate the governor. A leaf spring 230 mounted to the housing 176 by any suitable means engages a button 232 which is press fitted into the central bore of cam gear 222 to normally urge the latter axially toward partition 180, holding pin 228 against accelerating stop surface 227. However, when rotation of drive gear 170 and cam gear 222 is suddenly slowed, as by sudden slowing of the vehicle wheels upon the presence of a skid condition, the rotational inertia of shaft 202, especially due to the governor weights 208,

causes shaft 202 to turn ahead of gear 222, causing pin 228 to leave stop surface 227 and move to stop surface 225. This slides gear 222 on shaft 202 in a direction away from partition 180 to activate governor control.

It is important to note that solenoid spring 194 is stronger than governor spring 218, whereupon when solenoid 188 is not energized, plunger extension 192 bears against spool 220 to restrain same in the position illustrated in FIG. 4 wherein the rotary valve 126 is in the position illustrated in FIG. 6, namely, the position where atmosphere is free to pass through the valve to conduit 104 and then to chamber 98 and wherein at the same time suction from the intake manifold is blocked from communication with conduit 104. This is the normal operating condition of the system wherein the balanced pressure on opposite sides of actuator piston 84 enables spring 100 to position piston 84 and piston 68 as illustrated in FIG. 2. In this position, ball valve 64 is unseated to allow fluid pressure to pass directly to the brake line whereupon operation of the brakes is under direct control of brake pedal 12. This condition will remain so long as solenoid 188 (FIG. 4) is not energized. Energization of solenoid 188 is accomplished in the following manner. As previously explained, sudden slowing of gear 170 due to sudden slowing of the vehicle wheels as a result of a skid condition causes cam gear 222 (FIGS. 4, 5, 5A) to move axially on shaft 202 away from partition 180 against the action of spring 230. This movement is sufficient to cause actuation of a switch 234 suitably mounted in chamber 186. At the same time, the initial movement of the brake pedal and the resultant increase in fluid pressure in the system, causes hydraulic switch 18 (FIG. 2) to become closed. Hydraulic switch 18 is fully disclosed and described in copending U.S. patent application Ser. No. 203,075, filed June 18, 1962. Basically, switch 18 comprises piston 236 which has connected thereto at its upper end a metallic contact cup 238. Cup 238 is adapted to bridge contacts 240 and 242 but is normally maintained spaced therefrom by spring 244. An adjusting screw 246 is provided by means of which the tension of spring 244 may be varied, thus enabling the switch 18 to be preadjusted whereby a predetermined fluid pressure, e.g., 120–125 p.s.i., will cause piston 236 to move against the action of spring 244 sufficiently to bridge the contacts 240, 242. Actuation of switch 18 in the aforedescribed manner enables current to flow from battery 248 through contact 242, contact cup 238, and contact 240 to line 250 to the solenoid coil positive terminal 252 (FIG. 4). The voltage is applied through the solenoid coil 254 and then through line 256 to terminal 258 on switch 234. When switch 234 is open, the circuit to ground is not complete, and solenoid 188 will not be energized. However, upon actuation of switch 234 in the manner aforedescribed, the circuit to ground is completed through terminal 260, line 262 and ground connection 264. It is important to note that the circuit is such that although it is essential for hydraulic pressure switch 18 to be closed in order for solenoid 188 to be energized, closing of switch 18 alone is not sufficient, but rather switch 234 must also be actuated in order to complete the circuit. This is important because under normal operating conditions of the system sufficient pressure will normally be introduced to switch 18 by the foot pedal to actuate same, but it is essential that solenoid 188 not be energized under such normal conditions.

Energization of solenoid 188 accomplished by rightward movement of the cam gear (FIG. 5) closing switch 234, causes plunger 190, 192 to be retracted against the action of spring 194, thus freeing the shaft 200 and spool 220 for axial movement under the influence of the governor assembly. Therefore, the cam gear operating in response to the inertia of the governor and the solenoid operated plunger together can be regarded as an inertia responsive means for initiating the action of the governor.

As soon as the solenoid plunger has been retracted, due to the sudden slowing at the commencement of a skid condition, the governor assembly takes over control and, as it is stopped by stop surface 225 of the cam gear, spring 218 will cause the governor assembly to commence to collapse, thus moving shaft 200 and spool 220 axially to the left, when viewing FIG. 4. This movement of spool 220 causes corresponding rotary movement of valve 126, thus shutting off the passage of atmosphere to brake control conduit 104 and simultaneously effecting communication between vacuum passage 119 and conduit 104. This results in suction being introduced to chamber 98 which in turn overcomes the action of spring 100 to cause pistons 84 and 68 to move to the left, viewing FIG. 2. This movement of piston 68 enables ball valve 64 to close due to the action of spring 62, thus preventing the introduction of further fluid pressure to the system via brake pedal 12. Further leftward movement of piston 68 enables the fluid in lines 78 and 80 to back up, thereby relieving the pressure until the vehicle brakes begin turning, the duration that the suction is applied, and hence the degree of vacuum and the amount of pressure relief depending on the slipperiness of the road. As soon as the brake pressure is achieved, the vehicle wheels commence to rotate, whereby drive shaft 142 through the aforedescribed gearing on e again causes gears 170 and 222 to rotate, it being apparent that this [illegible]d rotation of shafts 200 and 202 will cause the governor assembly to once again turn, and due to the centrifugal force created, to retract shaft 200 and spool 220 to once again rotate valve 126 to the positions illustrated in Figs. 2 and 4, where once again atmosphere is connected to chamber 98, thus enabling brake pressure to be restored.

But, it is very important for my present invention that the solenoid is not de-energized and the governor returned to the restraint of the inertia-responsive device. Rather, the governor remains free to act. It leaves the valve 126 open to atmosphere 118 for an instant until the degree of vacuum is lessened enough to cause slight rightward movement of the piston 84 to cause the brake pressure to slightly exceed release pressure and a skid again commences. Again the governor is slowed by the drive train and again it opens the valve 126 to suction 118, and this cycle repeats again and again in an extremely rapid hunting or modulating action with the gas pressure in the conduit 104 never reaching either atmospheric or engine vacuum level, but rather modulates at intermediate levels. In dependent fashion, the hydraulic brake pressure modulates narrowly about the release pressure as a mean. Without returning to normal pressure. When the skid condition ends, the governor opens the valve continually to atmosphere and pistons 84 and 68 move fully to the right to once again unseat insolation valve 64 and restore normal brake operation. The faster the vehicle travels, the faster does this action occur, but the action is sufficntly fast down to speeds on the order of 10 m.p.h. and even lower, to rapidly pump and release the brakes. It is another very important aspect of the invention that the governor relinquishes control of the brake pressure before the vehicle stops.

In order to effectively carry out this modulating action, I have found it to be critical, in the preferred construction, that the solenoid 188 remain energized or that the plunger in some other way remains retracted until the vehicle slows to a predetermined slow speed, e.g., 5 or 10 m.p.h., depending on the vehicle. Since closing of switch 234 to complete the circuit to ground that energizes solenoid 188 is only momentary, holding means are provided for maintaining the solenoid energized until hydraulic pressure switch 18 opens, it being understood that switch 18 is preset to open only when fluid pressure in the system falls below the minimum value possible to achieve a skid condition, e.g., set to 85-90 p.s.i. produced by the governor at 5 or 10 m.p.h.

The aforesaid holding means for the solenoid comprise

a terminal 266 having a connection to ground as at 268, which terminal is engaged by plunger 190 when the latter is retracted in order to complete the circuit to ground independent of switch 234. Thus, although closing of switches 18 and 234 is essential in order toenergize solenoid 188, once the solenoid has been energized and its plunger retracted, switch 234 may be opened without de-energizing the solenoid, and the solenoid will remain energized until switch 18 opens.

As hereinbefore stated, it will be obvious that if switch 18 were not utilized, then there would be no means present for insuring de-energization of the solenoid once the condition has been reached where a skid is no longer possible, it being obvious that if the solenoid were still energized at such low speeds, the vehicle would be without effective brakes, since the governor assembly would not exert sufficient centrifugal force to overcome the action of spring 218 and move shaft 200 and spool 220 to the rightward position (FIG. 4) wherein valve 126 enables the brakes to be controlled by pedal 12.

It is possible for the reset device to take some other form, such as a pendulum device or a vehicle inertia memory system that can detect the slow speed of the vehicle, but brake pressure-sensitive devices, and particularly hydraulic-electric switches have important advantages in the system, particularly since they can respond whenever the skid condition has passed, regardless of the speed of the car, to enable normal operation, while offering low speed protection against the governor's cuttting off the brakes completely. The adjustment of the pressure level to which the device responds can be adjusted according to the wishes of the operator and the type of road conditions to be expected. But, in fact, a single setting below that at which skidding on ice occurs, which is determinable for any given vehicle, is advantageously employed to give anti-skid protection under virtually all conditions.

Any tendency for the unit 24 to erroneously sense a skid condition and actuate switch 234 in response to a rapid deceleration is overcome by use of properly calibrated weights 208 and a properly tensioned spring 230. In addition, I have found that an erroneous sensing is most likely to occur where there is a rapid acceleration of the drive shaft quickly followed by a rapid deceleraton. The frictional brake or drag that is resiliently exerted on spool 220 by plunger 190 helps to overcome this problem since this action tends to reduce the torque applied to shafts 200 and 202 by the governor assembly. It has been found, however, that at high speeds the frictional engagement between plunger 190 and spool 220 results in undue wear of the parts. Therefore, plunger 190 is assisted by spring 230, and the plunger is disengaged at high speeds. To this end, stop means 198 engages flange 196 so as to limit the outward movement of plunger 190 whereupon no engagement exists between plunger 190 and spool 220 at relatively high speeds. I have realized that the crucial problem of rapid acceleration followed by rapid deceleration occurs at low speeds in the low gear range with conventional automobiles, and I have, therefore, found that it is not necessary to have plunger 190 exert a frictional drag on spool 220 and shaft 200 at relatively high speeds.

Although the operation of my invention is thought to be clear from the foregoing description, a brief résumé of the operation will now be given. Under normal operating conditions, solenoid 188 is de-energized whereby the spring-loaded solenoid plunger maintains the skid-sensing and control unit inoperative. The de-energized solenoid further insures that rotary valve 126 is in the position wherein atmosphere is introduced to chamber 98. Since chamber 96 is also open to atmosphere, it follows that there is pressure balance on each side of large piston 84 thus enabling spring 160 to take over and move the piston 84 to the position illustrated in FIG. 2. In this position, piston 68, which is connected to piston 84 and moves therewith, engages ball valve 64 so as to unseat same against the action of spring 62. With ball valve 64 so opened, fluid pumped from master cylinder 14 by brake pedal 12 is free to pass directly to hydraulic lines 78 and 80 to actuate the vehicle brakes. The pressure-sensitive switch is adapted to be closed by a low level of brake pressure, e.g. 120–125 p.s.i., placing the anti-skid unit into readiness for operation.

Upon encountering a skid condition, i.e., a condition wherein the vehicle operator applies more brake pressure than is required for a particular road surface, the vehicle wheels slow suddenly. This causes corresponding deceleration of drive shaft 142, and, through suitable gearing, drive gear 170 and cam gear 222 also slow abruptly. The inertia of the governor assembly causes the gear 222 to move longitudinally by the cam surface aforedescribed, said longitudinal movement of gear 222 causing switch 234 to be closed which results in energization of solenoid 188 and the retraction of solenoid plunger 190. The second switch or terminal 266 is engaged upon retraction of the solenoid plunger to maintain the solenoid energized until switch 18 opens, even though switch 234 has now opened. Switch 18 is preset, e.g., at 85–90 p.s.i. to be opened at a brake pressure below that at which skids can occur, e.g., 100–150 p.s.i. for ice, whereupon solenoid 188 will remain energized until possibility of skidding no longer exists, and the solenoid is then de-energized to restore normal foot brake action before the governor has a chance ever to totally relieve the brakes.

Regarding the operation of the governor, once the solenoid 188 has been energized, the governor control unit takes over. More specifically, when a skid condition is approached, the sudden slowing of the vehicle wheels causes the governor assembly to collapse under the action of spring 218, thus moving shaft 200 and spool 220 to rotate valve 126, this movement now being permitted due to retraction of plunger 190. Rotation of valve 126 blocks the flow of atmosphere to chamber 98 and simultaneously allows the introduction of a vacuum to said chamber. The introduction of vacuum overcomes spring 100, thus resulting in movement of pistons 84 and 68 to the left, viewing FIG. 2. This movement closes isolation valve 64, thus preventing the application of further fluid pressure by brake pedal 12. As the valve remains in this position, the degree of vacuum increases, causing further leftward movement of piston 68 (FIG. 3), so that pressure in lines 78 and 80 and the brakes are relieved. A point is reached when the brakes are relieved sufficiently to allow the wheels to speed up from their slowed condition, the degree of relief required depending on the degree of slipperiness of the road. Then shafts 200 and 202 speed up once again, whereby the governor assembly retracts shaft 200 and spool 220 to once again rotate valve 126 to the position where atmosphere is introduced to chamber 98 and the piston 68 moves to the right and increases the brake pressure, until a part is reached where the wheels slow again and the cycle repeats. This action continues in a hunting manner so long as the vehicle is in an incipient skid, thus insuring that maximum brake pressure is continually applied to the wheel brakes short of pressure that would cause locking of the wheels. Once the brake pedal is released or the brake pressure in the system has dropped below the minimum possible value to create a skid, hydraulic pressure switch 18 breaks the circuit to solenoid 188, whereupon the latter becomes de-energized and the brakes once again become under complete control of the operator.

It is important to note that although loss of vacuum in the vehicle system will render the anti-skid means ineffective, said loss of vacuum will in no way affect normal operation of the vehicle brakes. Also, it is important to note that this system can be used with foot pedal controlled brakes of both the powered and nonpowered types.

While a spring-loaded fly ball type of governor has been illustrated, and is presently preferred, the broad

concept of the invention include the use of other governors, all rotary weight governors, such as gravity loaded fly ball, centrifugal shaft and inertial shaft governors, as well as governors that operate through electrical action.

The concepts are likewise applicable to other brake systems including air brakes in which valve 126 could directly modulate air brake pressure. Likewise it is possible to separate the governor from the valve, let the governor actuate a switch and let the valve be actuated by a solenoid responsive to the condition of the switch.

While use with engine vacuum is very efficient and inexpensive, and unlike previous systems I have proposed, does not cause the engine to run roughly due to loss of vacuum, the invention is applicable to other sources of pressure differential, incompressible fluid as well as gaseous fluid and numerous aspects of the invention are useful with nonfluid types of brake systems.

Referring now to FIGS. 7 and 8, a slightly different form of my invention is disclosed in which the relieving device is incorporated as an integral part of a power brake unit. The system generally comprises a brake pedal 300, a master cylinder 302, an accumulator 304, a safety valve 306, a hydraulic pressure switch 308, a power brake control unit 310, and an anti-skid unit 312.

Actuation of brake pedal 300 operates pistor 314 to pump fluid from cylinder 316 to conduit 318. Master cylinder 302 further comprises a fluid reservoir 320 communicating with cylinder 316 by means of passageway 322. A conduit 324 extends from reservoir 320, the function of which will hereinafter be described. Accumulator 304 is in communication with conduit 318 and comprises a spring loaded piston 326 and acts as a cushion and shock absorber for the fluid system where variations in fluid pressure occur.

During normal operation of the vehicle with which the power brake unit is associated, and with the brakes in their non-actuated position, the power brake unit 310 is in the position illustrated in FIG. 8. In this position, chambers 328 and 330 of cylinder 332 are both under vacuum, thereby equalizing the pressure on opposite sides of piston 334. This enables spring 336 to urge the piston 334 to the leftward position illustrated, said piston 334 being connected by means of rod 338 to a brake actuating pistion 340. Piston 340 is slidably mounted on cylinder 342, said cylinder having an outlet conduit 344 leading to safety valve 306. Conduit 344 is also in communication with hydraulic pressure switch 308.

A supply of hydraulic fluid is located in cylinder 342 whereupon movement of piston 334 against the action of spring 336, and corresponding movement of piston 340 will force fluid from cylinder 342 under pressure through conduit 344 and with valve 306 in the position illustrated in FIG. 8 to conduit 346 and then to hydraulic lines 348 which lead to the wheel cylinders (not shown) of the vehicle's braking system. Thus it will be seen that movement of pistons 334 and 340 to the right, viewing FIG. 8, will cause operation of the vehicle brakes.

As hereinbefore indicated, under normal operating conditions of the vehicle, and with the brakes not applied, piston 334 and 340 are in the position illustrated in FIG. 8 due to the fact that a vacuum exists in both chambers 328 and 330. More specifically, suction from the intake manifold 350 of the vehicle passes through conduit 352 into chamber 330 and then through passageway 354, through openings 356 in diaphragm 358 to conduit 360, through rotary valve 362 of the anti-skid unit 312, through conduit 364 and then to chamber 328. The control unit generally shown at 366 which forms a part of power brake unit 310 is of conventional form and comprises the aforesaid diaphragm 358, hydraulically movable by means of piston 368 to a closed position wherein the diaphragm engages the head seat 370 to block communication between passage 354 and conduit 360. The piston 368 also controls movement of poppet valve 372 normally maintained closed by spring 374. Spring 376 normally urges diaphragm 358 to its open or unseated position, as illustrated in FIG. 8. An atmosphere intake is located at 378, and introduced of atmosphere into the system is controlled by valve 372 in a manner now to be described.

As above indicated, with the parts in the positions illustrated in FIG. 8, the brakes of the vehicle are non-operative due to the fact that suction is free to circulate through conduit 352 to chamber 330 and then through control unit 366 to conduit 360, through rotary valve 362 to conduit 364 and then to chamber 328. With vacuum on both sides of piston 334, spring 336 maintains the piston 334 and the contacted brake actuating piston 340 in brake release position. Upon actuation of brake pedal 300 by the operator, pressurized fluid passes through conduit 318 to conduit 320. With the safety valve 306 in the position illustrated, flow of the fluid into the valve is blocked, and hence the brake fluid is forced to flow through conduit 380 into the brake control unit 366. None of the fluid from the pedal is allowed to reach the brake actuators at 348.

Introduction of the pressurized fluid to the brake control unit 366 causes actuation of piston 368 to move diaphragm 358 to its closed or seated position and to simultaneously open poppet valve 372 in a conventional manner. This movement blocks communication between chamber 330 and conduit 360 and simultaneously allows air to enter through inlet 378 through the now open valve 372 to conduit 360. The atmosphere circulates around conduit 360 through valve 362 to conduit 364 and then the chamber 328. The introduction of atmosphere pressure to chamber 328 created a pressure differential which overcomes spring 336 and moves piston 334 and piston 340 to brake actuating position. More specifically, the movement of piston 340 in cylinder 342 forces the brake fluid through conduit 344 and safety valve 306 to conduit 346 and hydraulic brake lines 348 to actuate the vehicle brakes in known fashion.

In order to insure that there is always an adequate supply of brake fluid in cylinder 342, conduit 324 connects fluid reservoir 320 to the cylinder 342, it being important to note that the inlet 382 to the cylinder 342 is located behind the forward portion of piston 340. Piston 340 is provided with a one-way valve 384 which enables fluid to pass through the piston from left to right, viewing FIG. 8, but which blocks flow of fluid through the piston from right to left. Thus, during operation of the piston 340, the suction created thereby in cylinder 342 sucks fluid through conduit 324 from reservoir 320 into cylinder 342 on the rear side of piston 340. During the return stroke of piston 340, the valve 384 allows fluid to pass through the piston, thereby insuring that there is always a full supply of fluid in the cylinder for brake action.

Since the structure and operation of hydraulic pressure switch 308 and anti-skid unit 312 are identical to that already described in connection with the first form of my invention, it is not thought that further description is necessary. It will be noted, however, that where the operator applies more brake pressure than is required for a particular road surface, the vehicle wheels will lock, whereby the same structure and operational sequence will cause rotary valve 362 to be moved to a position wherein conduit 360 becomes blocked and conduit 362A becomes open to allow suction from manifold 350 to be introduced through conduit 364 to chamber 328, whereupon spring 336 will once again take over to release the brakes. The same modulating action as aforedescribed will take place until hydraulic pressure switch 308 eventually renders the anti-skid inoperative.

In accordance with my invention it is crucial that the entire brake pressure be controlled by the anti-skid unit and that no fluid be conducted directly from the foot pedal to the wheels. Means are provided to insure that brake pressure will not be lost should there be a loss of suction. This is accomplished by safety valve 306. More specifically, conduit 362B is connected to the suction con-

duit 352 whereupon as long as proper suction exists in the system, piston 356 is maintained in the position illustrated in FIG. 8 against the action of spring 358. In this position, conduit 344 is in communication with conduit 345, but conduit 380 is blocked at its extremity adjacent the safety valve. Should suction be lost in the system, spring 358 will become effective to move piston 356 upwardly, thereby blocking the communication between conduits 344 and 346 and at the same time providing communication between conduit 380 and conduit 345. With the safety valve in this position, the brakes are directly controlled by operation of pedal 300 since the entire power unit is now bypassed.

A one-way check valve 390 is located in conduit 352 adjacent the intake manifold 350 since without same backfire of the vehicle engine would cause an explosive pressure to be introduced to the power unit, which would be undesirable for obvious reasons.

Referring once again to FIG. 4, it will be noted that should the vehicle move in reverse, the direction of rotation of drive shaft 142 will likewise be in reverse, thus causing reverse rotation of flexible shaft 164 and shaft 168. Reverse rotation of shaft 168 will, in turn, cause reverse rotation of drive gear 170 and cam gear 222, causing pin 228 to leave stop surface 227 and move toward stop surface 225, which, as previously described, causes gear 222 to slide on shaft 202 in a direction away from partition 180 to close switch 234 and thus activate the governor control and anti-skid means. Even though it is possible that very slow reverse movement of the vehicle might not be sufficient to cause movement of cam gear 222 to activate the governor control, the fact remains that there is always the possibility and even likelihood that reverse movement of the vehicle will activate the governor control even though no skid condition exists. Since there is no practical need for any anti-skid means when the vehicle is moving in reverse, and since the activation of these means when not needed might even result in the loss of some stopping distance, it has been found desirable to provide means for rendering the governor control and anti-skid ineffective and inoperative when the vehicle moves in reverse.

Referring to FIGS. 9 through 12, a somewhat modified anti-skid unit is shown generally at 400. Unit 400 is basically the same as skid-sensing and control means 24 hereinbefore described except that a reverse cut-off switch has been incorporated in the unit to prevent energization of solenoid 188 when the vehicle drive shaft is in reverse and flexible shaft 164 correspondingly rotates in a reverse direction. The reverse cut-off switch, shown generally at 402 (FIG. 10) comprises a shaft 404 slidably coupled to shaft 164 as at 406, in the same manner that shaft 168 was drivingly engaged to flexible shaft 164. Shaft 404, which is constructed of any electrically conductive metal, is journalled in bore 408, said bore having an enlarged end portion 410, it being understood that the larger diameter of portion 410 results in the presence of an annular shoulder 412 at the extremity of bore 404. Slidably mounted in portion 410 is a contact button 414, said button being resiliently urged toward bore 408 by means of spring 416. The opposite end of spring 416 resiliently engages a contact cup 418 having a terminal 420 extending radially therefrom.

Adjacent its opposite extremity, shaft 404 has fixedly mounted thereon the afore-described drive gear 170, said gear meshing with cam gear 222, all in the manner hereinbefore described. A spring 422 engages washer 424 to normally urge shaft 404 to the left when viewing FIGS. 9 and 10, thus causing the end of shaft 404 to extend into enlarged portion 410 and, therefore, into engagement with contact button 414, as illustrated in FIG. 9. Loosely mounted on shaft 404 is a reverse switch cam 426 comprising a circular plate 428 having a spiral cam portion 430 mounted thereon, and further having an outwardly extending pin 432 located adjacent the end of cam track 430. A cam pin 434 extends radially from shaft 404 and is adapted to cooperate with the spiral cam track 430 to determine the axial position of shaft 404. Expressed differently, when shaft 404 is rotating in a counterclockwise direction when viewing FIG. 12, cam pin 434 will engage the stop pin 432 at the low point on the spiral track 430. With the parts in this position, shaft 404 will assume the position illustrated in FIG. 9 wherein the extremity of the shaft is in engagement with contact button 412, it being understood that spring 422 is somewhat stronger than spring 416, whereupon the extremity of shaft 404 will actually enter into enlarged portion 410. Upon clockwise rotation of shaft 404, pin 434 will ride up the spiral cam track 430 until it reaches the high end of said track and engages the opposite side of stop pin 432, as illustrated in FIG. 10. In this position, shaft 404 has been moved to the right (when viewing FIG. 10) against the action of spring 422 sufficiently for the end of said shaft to become completely withdrawn from enlarged portion 410. Since the annular shoulder 412 prevents the contact button 414 from moving out of enlarged portion 410, it follows that the end of shaft 404 becomes disengaged from contact button 414, as clearly illustrated in FIG. 10. Also mounted on shaft 404 is a washer 436 having a radially extending contact arm 438, the purpose of which will hereinafter be described. As will be seen most clearly in FIG. 10, the washer 436 is mounted on shaft 404 adjacent the flat rear surface of cam 426, it being understood that shaft 404 may freely rotate within washer 436. A cam thrust washer 440 comprising a peripheral ring portion 442 and bent marginal lugs 444 resiliently engages the opposite face 428 of cam 426, whereupon to resiliently urge cam 426 into engagement with washer 436 and at the same time to impart a sufficient frictional drag on cam 426 so that movement of shaft 404 will not impart corresponding rotary movement to the cam until pin 434 moves into engagement with stop pin 432. The thrust washer 440 has a radially extending spring arm 446 that is secured to the housing 176 by any suitable means, such as by having the end of the spring arm clamped between adjacent portions of the housing, as illustrated in FIGS. 9 and 10.

The operation of reverse switch 402 is as follows. Assuming that shaft 404 is being rotated in a counterclockwise direction, which for purposes of this description correspods to forward movement of the vehicle, the switch will, as afore stated, assume the position illustrated in FIG. 9 wherein shaft 404 is making engagement with contact button 412. With the parts in this position, and assuming that a skid condition commences to exist, causing cam gear 222 to close switch 234, current will flow through line 262 to terminal 420 through contact cup 418, spring 416, contact button 414, and then to shaft 404, which as previously described, is in engagement with contact button 414. Shaft 404 is in turn in electrical engagement with washer 436 and its radially extending terminal 438, both directly and through the engagement of pin 434, which in turn engages cam 426, which in turn bears against washer 436. Thus, the flow of current is through terminal arm 438 and then to line 262A, which connects to ground as at 264. This, along with line 256, completes the circuit to solenoid 188 in order to energize same and render the governor control effective, all in a manner hereinbefore described. On the other hand, should the vehicle move into reverse, thus resulting in clockwise rotation of shaft 404, cam 426 will cause the shaft 404 to retract and become disengaged from contact button 414, as illustrated in FIG. 10, thus interrupting the connection to ground and preventing energization of solenoid 188. It will thus be seen that whenever the vehicle and its drive shaft move in reverse, the reverse cut-off switch 402 will automatically prevent the skid sensing and control means and the governor control from coming into play. Thus, the vehicle brakes

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will always be under normal and direct control by the operator whenever the vehicle is moving in reverse.

While there is shown and described herein certain specific structure embodying the invention, it will be manifest to those skilled in the art that various modifications and rearrangements of the parts may be made without departing from the spirit and scope of the underlying inventive concept and that the same is not limited to the particular forms herein shown and described except insofar as indicated by the scope of the appended claims.

I claim:

1. A control system for operating a vehicle wheel brake-relieving device for the purpose of preventing skids, said control system comprising the combination of an inertia-responsive means adapted to respond to a sudden decrease in the speed of rotation of the wheel and thereupon to initiate the operation of said relieving device, a rotary governor means adapted to rotate dependently with the speed of the wheel and adapted after said operation is initiated by said inertia-responsive means to correspondingly vary the condition of said relieving device, to decrease and increase the brake pressure dependently with decrease and increase in the speed of the wheel, and a reset means adapted to respond to a predetermined, decreased speed of the vehicle, to terminate the operation of said relieving device, and reset said inertia-responsive means so that it may again initiate said operation.

2. The system of claim 1 wherein said reset means includes a pressure-responsive means, adapted to respond to a predetermined decreased brake pressure level, said governor means adapted to produce said level when said vehicle reaches said predetermined decreased speed, said predetermined brake pressure level being below the lowest level which the governor means can produce when said vehicle travels at speeds in excess of said predetermined speed.

3. The system of claim 2 wherein said inertia-responsive means that initiates the operation of said relieving device is adapted to operate with electrical power, a power source for said inertia-responsive means, said pressure-sensitive means being adapted to switch off said power source from said inertia-responsive means when said brake pressure is at or below said predetermined level, to prevent the operation of said relieving device, and said pressure-sensitive means adapted to switch on said power source to said inertia-responsive device when said brake pressure is above said predetermined level, so that said control system is placed into condition for operation whenever the brakes are operated.

4. The system of claim 2 wherein said reset means comprises a pressure-sensitive switch set to respond to a predetermined brake pressure below that brake pressure which can cause skidding of said wheel on ice.

5. The system of claim 1 wherein said inertia-responsive means includes a control member associated with said governor, said member normally urged to a first state to prevent said governor from operating said relieving device, a solenoid associated with said governor control member, adapted when energized to move said control member to an inactive state to free said governor to operate, a switch-actuating member adapted to be moved under the influence of the rotary inertia of said governor when said wheel changes speed suddenly, said switch-actuating member adapted to close and open a switch depending on whether said wheel is slowing down or speeding up, said switch adapted, when closed, to energize said solenoid, and wherein a holding means is provided to retain said solenoid energized when said switch thereafter opens, and said reset means is adapted to de-energize said solenoid, to allow said governor control member to return to said first state.

6. The system of claim 5 wherein said governor control member comprises a plunger disposed within the coil of said solenoid, and said holding means comprises a second switch associated with said solenoid, positioned to be engage by said plunger when in the inactive position, said switch adapted to complete the circuit for said solenoid independent of said first switch, thereby serving to maintain said solenoid energized when said first switch opens.

7. A skid control system for a pressure-operated vehicle wheel brake, the pressure at said brake normally controlled by the driver of the vehicle, said control system comprising an anti-skid means adapted, on the occurrence of a skid condition during braking, to repeatedly decrease the pressure at the vehicle wheel brake dependently with decrease in the speed of the wheel, and a pressure-sensitive switch set to detect the decrease of the pressure at the wheel brake to a predetermined minimum level, and thereupon remove the influence of said anti-skid means and return to normal control the pressure at said wheel brake.

8. A control apparatus adapted to control braking of a vehicle wheel during skid conditions comprising a rotary weight governor device adapted to rotate dependently with said wheel, said governor device having a member adapted to move between a first position and a second position dependently with changes in the speed of rotation of said governor device, a brake pressure control means responsive to said movement to relieve said brakes when said member is moved toward the first position with decrease in wheel speed and restore the pressure of said brakes when said member is moved toward the second position with increase in wheel speed, a restraining means operable between an active state at which it maintains said member of said governor in said second position at which the brake pressure is under the control of vehicle driver, and an inactive state at which it allows said governor member to move, and means adapted to place said restraining means in said inactive state when a wheel skid condition first occurs, the improvement wherein means are provided for maintaining said restraining means steadily in its inactive state, once said inactive state is reached, to allow said governor to move said member repeatedly between said positions, dependently upon the speeds of the wheel, whereby the brake pressure can be controlled at a level near but below the pressure at which locking of the wheel occurs, and reset means adapted to reactivate said restraining means, when the vehicle slows, before it has stopped.

9. The control apparatus of claim 8 including a rotary valve associated with said governor for controlling said brake pressure through a gas pressure conduit, said rotary valve having a rotary member adapted to vary the pressure in said conduit dependently with charges in its rotary position, a projection extending from said rotary member and mechanically connected to said movable member of said governor to cause said rotary member to move dependently with said governor member.

10. The control apparatus of claim 8 wherein said rotary weight governor is of the fly ball type having a rotatable shaft free to move axially in response to speed changes, and said restraining means comprises a movable plunger, in its active condition urged against said shaft in a manner to prevent outward movement of the latter, said inertia-responsive means adapted to respond to the rotary inertia of said governor when said wheel slows, and a stop member limiting the movement of said movable plunger so as to prevent engagement with said shaft at relatively high rotational speeds due to axial movement of said shaft inwardly to a position out of the range of said movable plunger whereby frictional drag is applied by said movable plunger to said shaft only at relatively low rotational speeds.

11. In an anti-skid apparatus for use in a brake system employing means normally controlling brake pressure to a vehicle wheel brake through a hydraulic pressure line, a separate relieving device adapted to relieve the brake pressure at the wheel brake, and a control apparatus adapted to control said relieving device, said relieving device comprising an isolation valve adapted normally to

be open but able to seal said wheel brake from the effect of said normal control means and means adapted to relieve the fluid pressure at said wheel brake, said isolation valve including a movable member adapted to seat to prevent pressure communication to said wheel brake, and said relieving means comprising a piston disposed in a hydraulic cylinder, said cylinder in communication with said hydraulic line at a location between said wheel brake and said isolation valve, said piston being adapted to directly operate said isolation valve, said piston having a projection adapted to engage said movable member of said valve, to position said movable member in an unseated position and to allow said member to seat dependent upon the position of said piston, said piston being movable beyond the position at which said member is allowed to seat, to relieve pressure at said wheel brake when said isolation valve is in sealing position, said control device adapted to regulate the position of said piston variably in accordance with skid conditions alternately applying and relieving the brake pressure at the wheel brake while said movable member is seated, the improvement wherein said piston is biased by a spring to position said movable member in the unseated position, in a manner to perform a pressure limiting function, said spring having a predetermined effective force that is less than the maximum force that can be applied oppositely on said member by said normal hydraulic pressure, whereby, when said maximum hydraulic pressure is applied, said spring can yield even when skid conditions do not exist, to allow said movable member to seat, to prevent said maximum pressure from being communicated to said wheel brake thereby limiting the amount of brake pressure that must be relieved when a skid condition occurs.

12. In an anti-skid system adapted to relieve fluid pressure applied by an operator to the brake for a vehicle wheel on the occurrence of skid conditions, said anti-skid system including a vehicle wheel brake-relieving device, control means adapted to respond to a sudden decrease in the speed of rotation of the wheel and thereupon to initiate the operation of said brake-relieving device, and means adapted after the initiation of the operation of said brake-relieving device to correspondingly vary the condition of said brake-relieving device to decrease and increase the fluid pressure applied to the brake dependently with decrease and increase in the speed of the wheel, the improvement comprising a resiliently loaded fluid pressure limiting device connected to the fluid pressure line and constructed and arranged to limit the pressure normally applied by the operator to the wheel brake to a predetermined maximum level, said level being less than the full pressure applicable by the operator, whereby, as a skid condition occurs when the operator jams on his brakes to the full extent, said anti-skid system is able to stop the skid without having to relieve the full pressure applied by the operator.

13. An anti-skid unit for pressure actuated vehicle brakes comprising a pair of aligned shafts, means connecting said shafts for rotation as a unit with one of said shafts axially movable with respect to the other, governor means connected to said shafts, said governor means being adapted to cause movement of said axially movable shaft in response to variations in rotational speed of said shafts, drive means operative in response to rotation of the vehicle wheels for imparting corresponding rotation to said shafts, valve means movable between a first position wherein said vehicle brakes may be applied and a second position wherein said brakes are automatically released, means controlling movement of said valve means in response to axial movement of said axially movable shaft, a normally deenergized solenoid having a plunger that maintains said axially movable shaft against axial movement and in a position wherein said valve means is in its said first position, and means for energizing said solenoid to retract said plunger and release said axially movable shaft for axial movement under the control of said governor means, said energizing means comprising a first switch, cam means automatically actuating said first switch in response to sudden slowing of said vehicle wheels and said drive means, and a second switch automatically actuated by the solenoid plunger when the latter is retracted to maintain the solenoid energized after said first switch opens.

14. The anti-skid unit of claim 13 further characterized in that said energizing means comprises a third switch, said third switch being closed by the presence of fluid pressure in the system, said solenoid remaining energized until fluid pressure in the system drops below the predetermined value to open said third switch.

15. The anti-skid unit of claim 13 further characterized in that resilient means are provided for urging said solenoid plunger into frictional engagement with said axially movable shaft when the solenoid is deenergized, thereby imparting a frictional drag to said axially movable shaft during rotation thereof, and stop means limiting the extent of movement of said plunger toward said shaft so as to prevent engagement therewith at relatively high rotational speeds due to axial movement of said shaft to a position out of the range of said plunger by said governor means.

16. A control system for operating a vehicle wheel brake-relieving device for the purpose of preventing skids, said control system comprising the combination of an inertia-responsive means adapted to respond to a sudden decrease in the speed of rotation of the wheel and thereupon to initiate the operation of said relieving device, a rotary governor means adapted to rotate dependently with the speed of the wheel and adapted after said operation is initiated by said inertia-responsive means to correspondingly vary the condition of said relieving device, to decrease and increase the brake pressure dependently with decrease and increase in the speed of the wheel, a reset means adapted to respond to a predetermined, decreased speed of the vehicle, to terminate the operation of said relieving device, and reset said inertia-responsive means so that it may again initiate said operation, and means responsive to reverse rotation of the vehicle drive shaft for preventing operation of said control system.

17. The system of claim 16 further characterized in that said last-mentioned means comprises a switch that is automatically closed when the vehicle drive shaft is rotating in a forward direction and which automatically opens when the vehicle drive shaft rotates in a reverse direction.

18. A skid control system for a pressure-operated vehicle wheel brake, the pressure at said brake normally controlled by the driver of the vehicle, said control system comprising an anti-skid means adapted, on the occurrence of a skid condition during braking, to remove control of the brake pressure by the operator and thereafter repeatedly decrease the pressure at the vehicle wheel brake dependently with decrease in the speed of the wheel, and reset means adapted to a predetermined, decreased speed of the vehicle, to terminate the operation of said anti-skid means and return control of the brake pressure to the operator and switch means controlled by the direction of rotation of the vehicle drive shaft for preventing operation of said control system when the drive shaft is rotating in reverse.

19. The system of claim 18 further characterized in that said switch means comprises a rotatable shaft drivingly interconnected to the vehicle drive shaft, a contact button adapted to resiliently engage one end of the rotatable shaft, said rotatable shaft being axially movable between a first position wherein it is in engagement with said button and a second position wherein it is disengaged therefrom, and cam means associated with said rotatable shaft for automatically causing axial movement thereof to said first position when the vehicle drive is in forward, and for automatically causing movement thereof to said second position when the vehicle drive is in reverse.

20. A vehicle wheel anti-skid system for use with a

vehicle brake system having a fluid source, a vehicle wheel brake operating actuator cylinder in fluid communication with said fluid source, and operator controlled means for applying fluid pressure to said vehicle wheel brake operating actuator cylinder, said vehicle wheel anti-skid system comprising means for sensing a predetermined deceleration of the vehicle wheel, means for removing from the operator control of the amount of the fluid pressure applied to the vehicle wheel brake operating actuator cylinder upon actuation of said sensing means, and means actuated by said sensing means for alternately applying and relieving the fluid pressure applied to the vehicle wheel brake operating actuator cylinder while control of the amount of fluid pressure applied to the vehicle wheel brake operating actuator cylinder is removed from the operator.

21. A vehicle wheel anti-skid system as in claim 20 including means for returning to the operator control of the amount of the fluid pressure applied to the vehicle wheel brake operating actuator cylinder under a predetermined set of vehicle operating conditions.

22. In a vehicle brake system, in combination, a fluid source, a vehicle wheel brake operating actuator cylinder in fluid communication with said fluid source, operator controlled means for applying pressure to said vehicle wheel brake operating actuator cylinder, a vehicle wheel anti-skid system interposed between said vehicle wheel brake operating actuator cylinder and said operator controlled means, said vehicle wheel anti-skid system including means for sensing a predetermined deceleration of the vehicle wheel, means for removing from the operator control of the amount of the fluid pressure applied to the vehicle wheel brake operating actuator cylinder upon actuation of said sensing means, means for alternately applying and relieving the fluid pressure applied to the vehicle wheel brake operating actuator cylinder while control of the amount of the fluid pressure applied to the vehicle wheel brake operating actuator cylinder is removed from the operator and means for returning to the operator control of the amount of the fluid pressure applied to the vehicle wheel brake operating cylinder

under a predetermined set of vehicle operating conditions.

23. For use in an automotive brake system employing a brake pedal and master cylinder to normally control brake pressure at a vehicle wheel brake, sensing means to sense a skid condition, control means responsive to said sensing means and employing vacuum produced by said vehicle as a power source, said control means adapted to relieve the pressure of the brakes, and means responsive to loss of said vacuum adapted to place said master cylinder in direct communication with said wheel brake, wherein said relieving means is incorporated in a power unit adapted to apply brake pressure in response to said brake pedal, said power unit constructed to prevent any direct communication between the brake pedal pressure and the wheel brakes when said engine is operating, and a two-position valve adapted at one position to connect said brake pedal pressure to control said power unit and connect said power unit with said wheel brake, and at the other position to connect said brake pedal pressure directly with said wheel brake, by-passing said power unit, means causing the vacuum of said engine to maintain said valve in said first position, and means responsive to loss of said vacuum to move said valve to said second position.

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[PX 10, p 1]

PLAINTIFF'S EXHIBIT 10

July 11, 1967 F. A. PERRINO 3,330,113

POWER BRAKES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 1

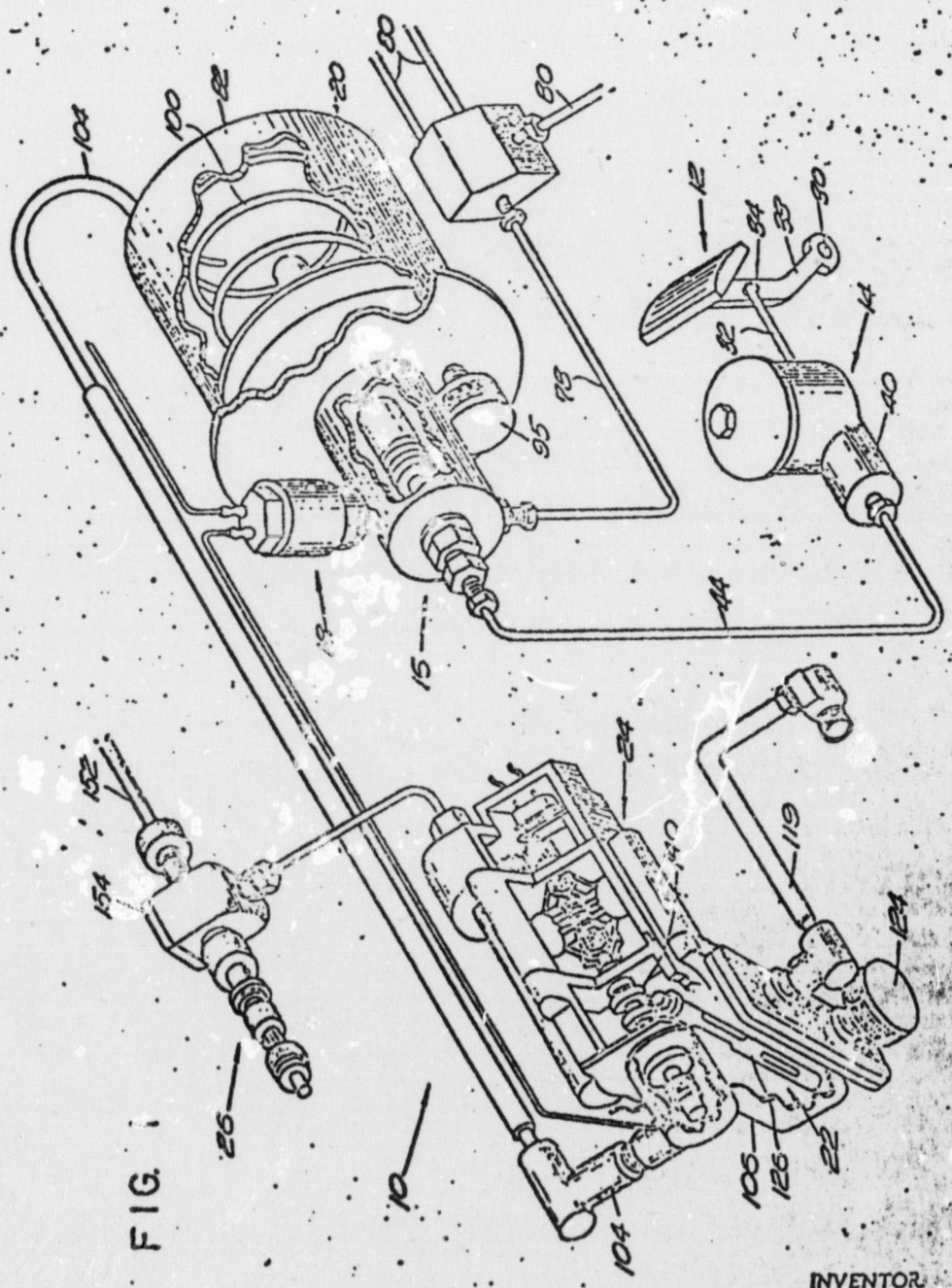


FIG. 1

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POWER BRAKES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 2

FIG. 2

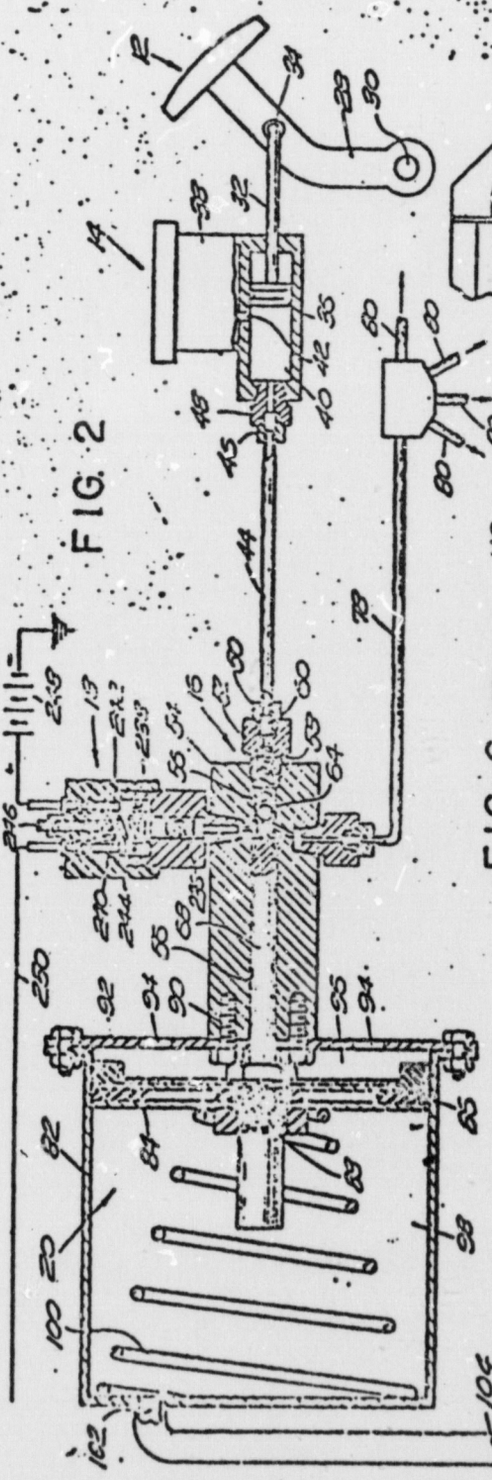


FIG. 5

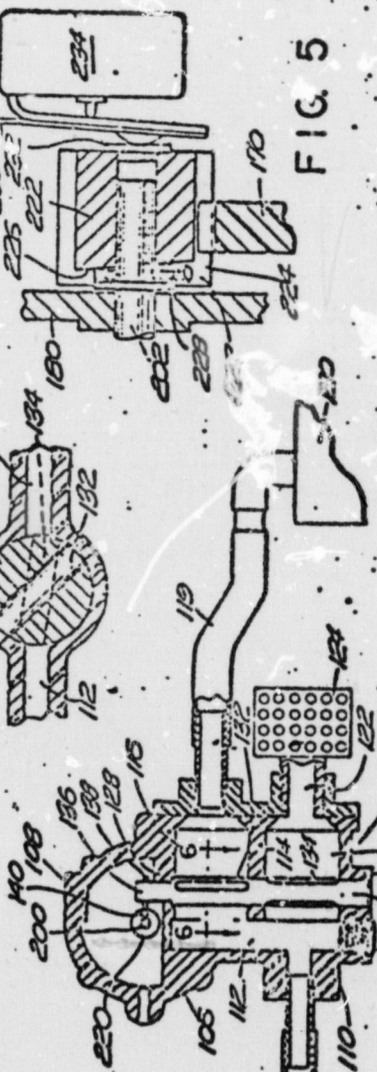


FIG. 6

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POWER BRAKES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 3

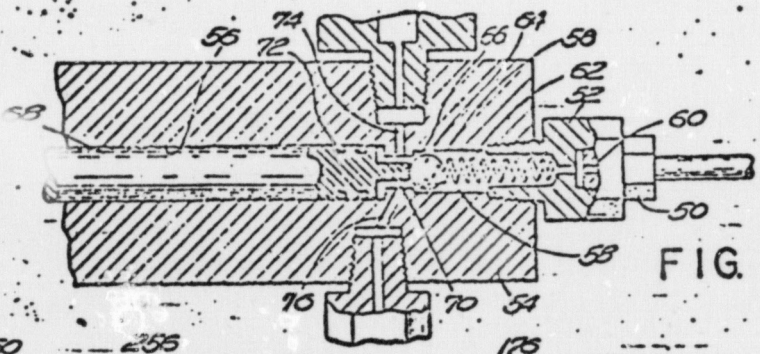


FIG. 3

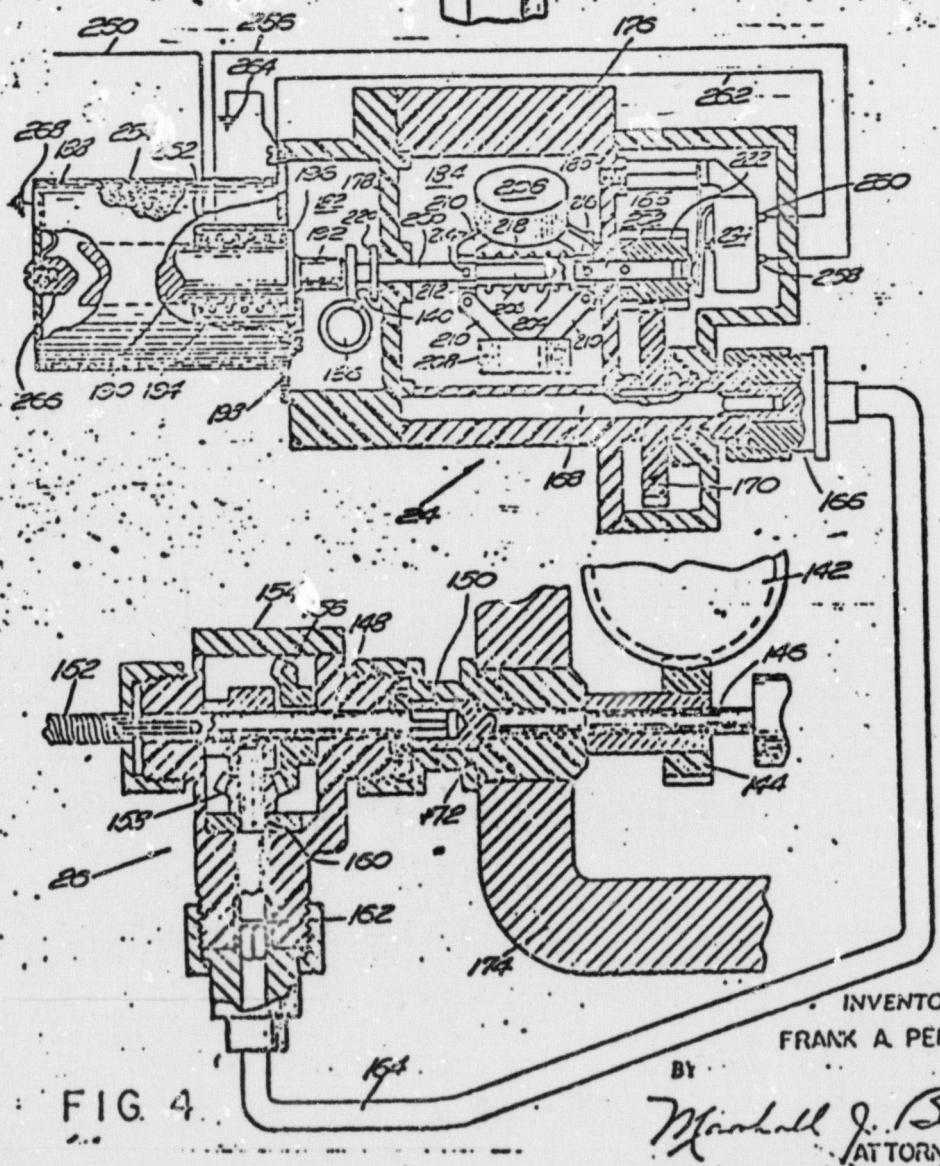


FIG. 4

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Original Filed Nov. 2, 1964 7 Sheets-Sheet 4

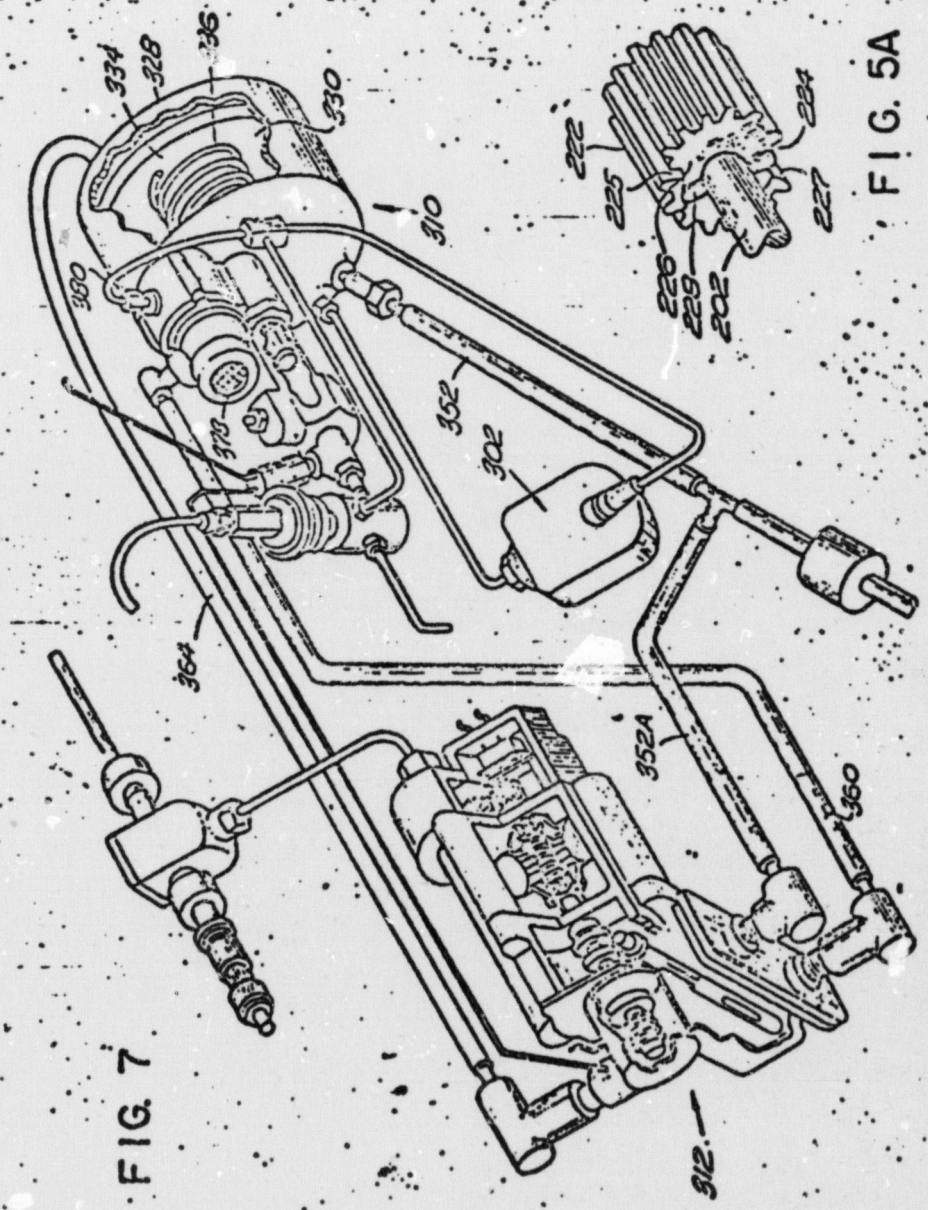


FIG. 5A

FIG. 7

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POWER BRAKES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 5

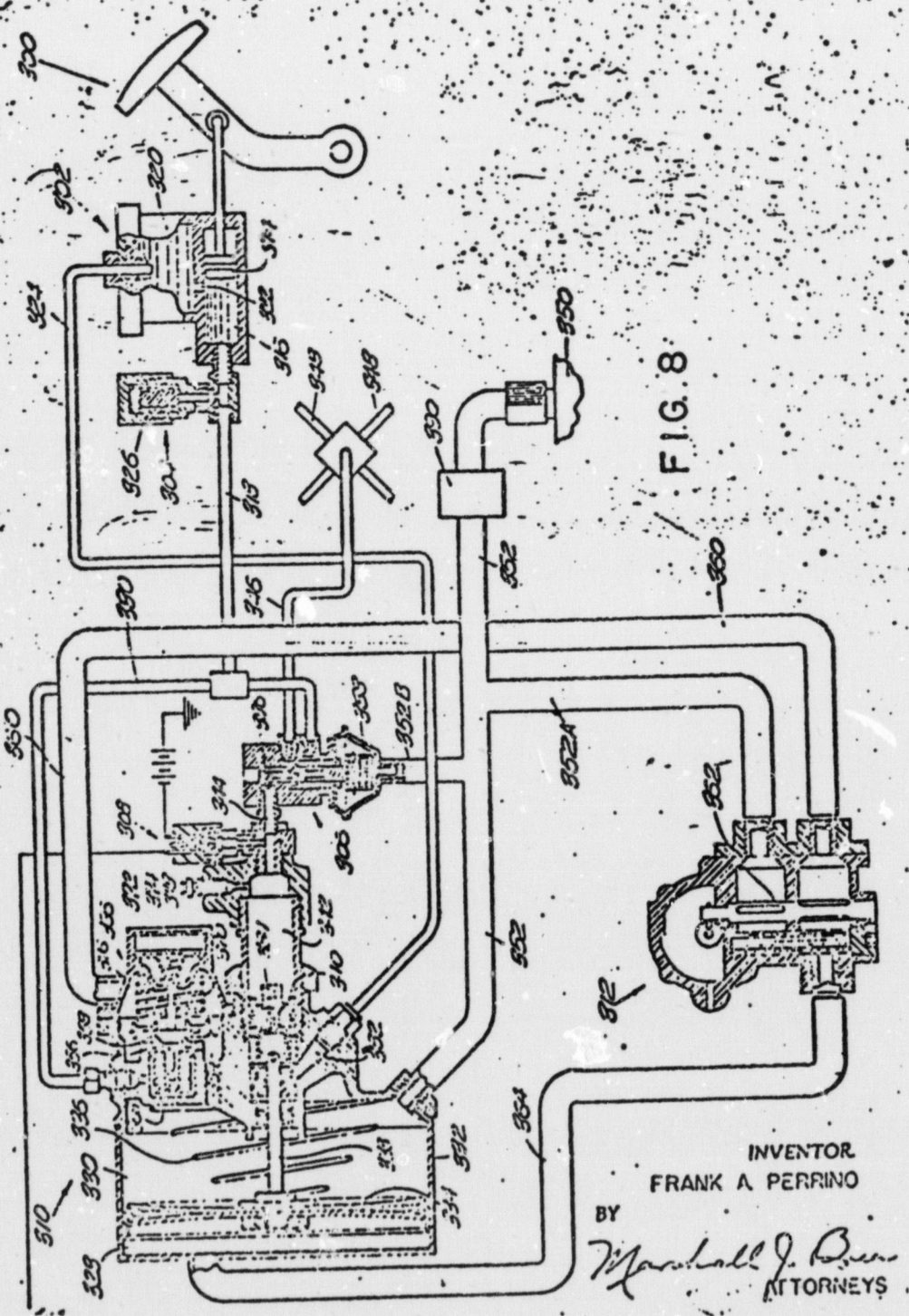


FIG. 8

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[PX 10, p 6]

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POWER BRAKES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 6

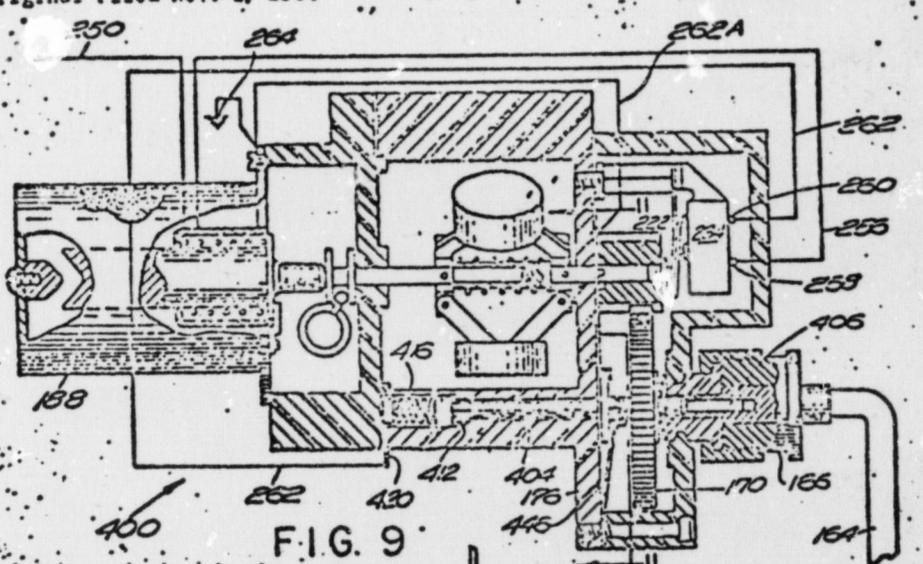


FIG. 9

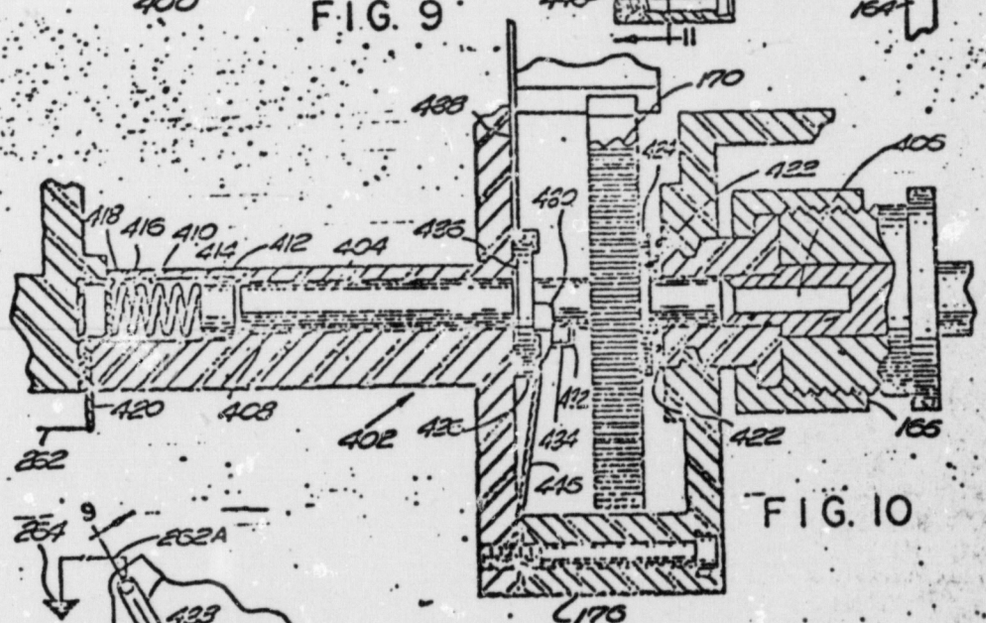


FIG. 10

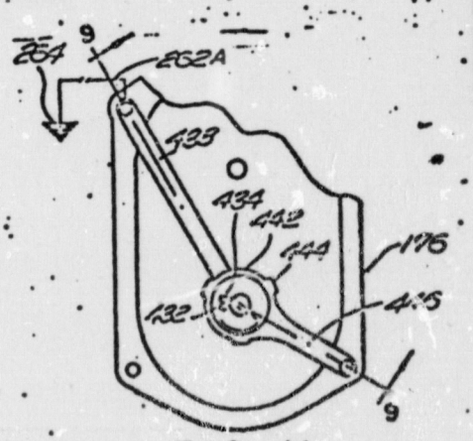


FIG. 11

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POWER BRAKES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 7

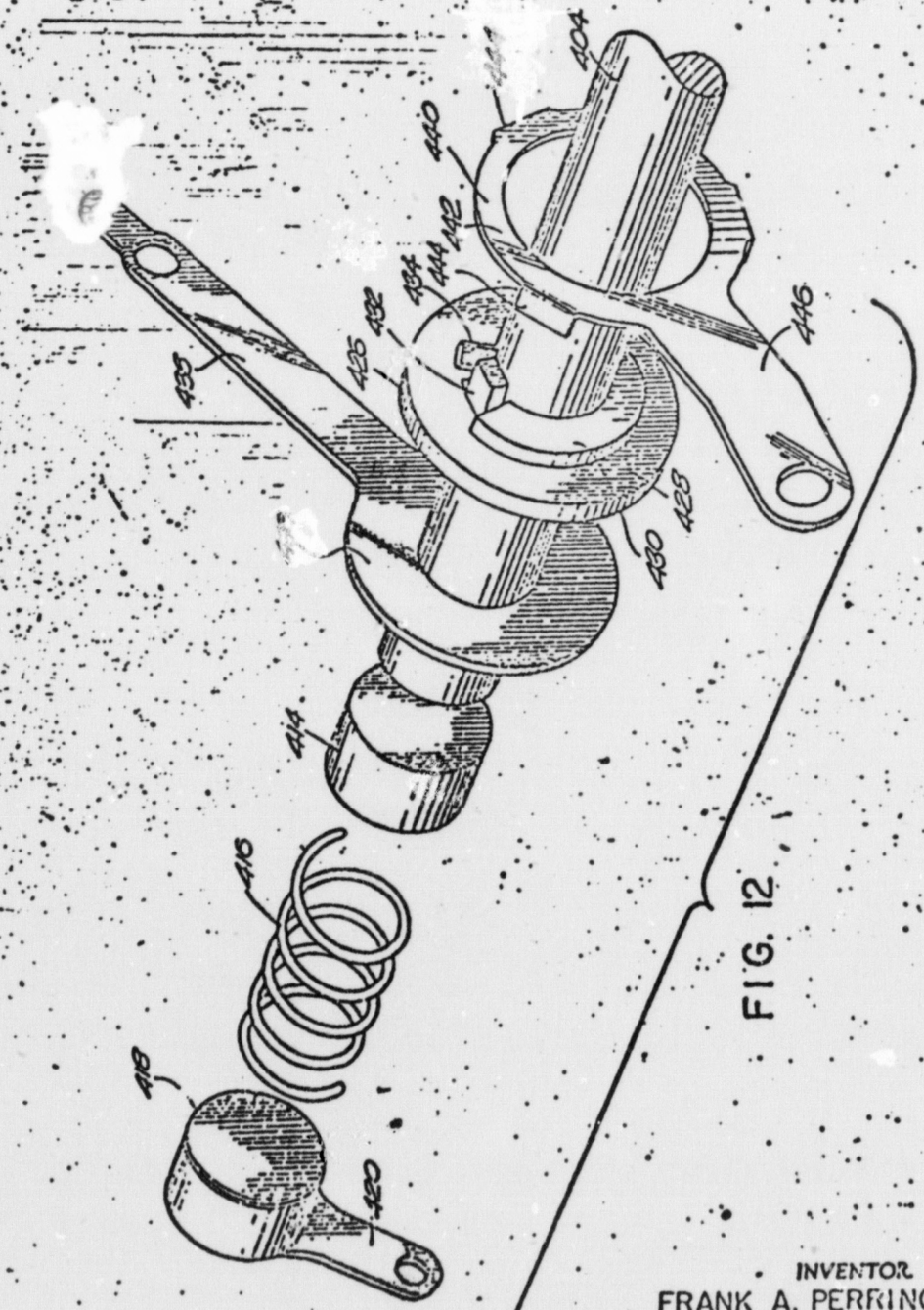


FIG. 12

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[PX 10, p 8]

United States Patent Office 3,330,113 Patented July 11, 1967

3,330,113

POWER BRAKES

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Original application Nov. 2, 1964, Ser. No. 409,350. Divided and this application Aug. 5, 1966, Ser. No. 581,675

9 Claims. (Cl. 60—54.5)

This is a division of application Ser. No. 409,350, filed on Nov. 2, 1964, and entitled Braking System and Anti-Skid Means Therefor.

The present invention relates generally to braking systems for automotive vehicles and the like and is particularly concerned with the provision of novel and improved anti-skid means.

It is well known that a skid condition may be encountered by a moving vehicle where the brakes of the vehicle are applied so hard that the vehicle wheels tend to lock. Quite obviously, other factors affect the commencement and existence of a skid condition, such as the condition of the surface on which the vehicle is moving, the speed at which the vehicle is traveling, etc. However, it is possible for a vehicle to commence a skid even where the surface on which the vehicle is traveling is not wet or icy and even where the vehicle is traveling at moderate speeds, if the vehicle brakes are applied too hard. Since all control of a vehicle is lost while the vehicle is in a skid, thus increasing the danger of serious accident and personal injury, it is obviously highly desirable to prevent or reduce skidding while at the same time enabling the vehicle to come to a complete stop as quickly as possible.

It is therefore a primary object of the present invention to provide an improved braking system wherein operation of the brakes is under control of the vehicle operator until a skid condition commences to exist, at which time anti-skid means automatically take over complete control of the brakes and maintain said control until danger of skidding no longer exists, whereupon the brakes are returned to normal control by the operator.

Another object of my invention is the provision of efficient and sensitive control means for the various operations of the systems, which have usefulness in other types of systems as well.

Another object of my invention is the provision of novel and improved anti-skid means equally adaptable to so-called power brakes, as well as nonpower brakes.

A further object of my invention is the provision of anti-skid means of the character described wherein the vacuum system of the vehicle is utilized, but wherein the loss of vacuum in no way interferes with control of the vehicle brakes by the operator.

A further object of my invention is the provision of anti-skid means that is readily and easily adaptable to existing vehicles.

Another object is the provision of a novel and improved reverse cut-off switch that automatically renders the anti-skid means inoperative when the vehicle moves in reverse.

Still another object of my invention is the provision of an anti-skid system for automotive vehicles and the like having means which are durable and effective in operation and which are feasible and practical both from an economic and manufacturing standpoint.

Other objects, features and advantages of my invention will become apparent as the description thereof proceeds when considered in connection with the accompanying illustrative drawings.

In the drawings which illustrate the best mode presently contemplated for carrying out the present invention:

FIG. 1 is a diagrammatic view, in perspective, of the

assembly utilized in my nonpower unit, with portions being broken away for purposes of illustration;

FIG. 2 is an elevational view, in section, of a portion of the overall assembly illustrated in FIG. 1;

FIG. 3 is an enlarged fragmentary sectional elevation of a portion of the brake-actuating means illustrated in FIG. 1;

FIG. 4 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid unit and the drive means therefor;

FIG. 5 is an enlarged sectional elevation of the cam gear and solenoid-actuating switch which form a part of the anti-skid unit;

FIG. 5A is an enlarged fragmentary perspective view of the cam gear;

FIG. 6 is an enlarged section taken on line 6—6 of FIG. 2;

FIG. 7 is a diagrammatic view, in perspective, of my anti-skid means in combination with a power brake unit, portions being broken away for purposes of illustration;

FIG. 8 is an elevational view, in section, of the power brake assembly illustrated in FIG. 7;

FIG. 9 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid means and the reverse cut-off switch in combination therewith;

FIG. 10 is an enlarged fragmentary sectional elevation of the reverse cut-off switch per se;

FIG. 11 is a fragmentary section taken on line 11—11 of FIG. 9; and

FIG. 12 is an enlarged exploded perspective view of the reverse cut-off switch per se.

According to one aspect of my invention, a sensing means for sensing when a skid condition begins is caused to activate a wheel speed-sensitive device that takes over control of the brake pressure until the skid condition ends. Advantageously, the sensing means is an inertia-responsive means that senses a sudden change in the speed of rotation of a wheel at the commencement of skid conditions. This inertia-responsive device does not control the brake pressure itself by repeated action during the skid condition, as has been attempted by the prior art. Instead the control of the brake pressure is taken over completely by a governor driven dependently with the wheel, advantageously a mechanical governor. The governor then controls the brake pressure in a hunting or modulating manner, for an extended period, until such control of the brakes is ended by a reset means that responds to slowing of the vehicle to a predetermined speed. Preferably, the reset means is a brake-pressure-sensitive device which senses reduction of brake pressure to a predetermined level which the governor can produce only when it slows to the predetermined speed.

This separation of the initiating and controlling functions is an extremely important feature. The initiating device must necessarily have a restraint to prevent operation when the vehicle merely decelerates normally. I have realized that such a restraint prevents the device from being able to effectively regulate the brake pressure due to lack of sensitivity. The governor, after it takes over control, is free of such restraint and therefore can have much greater sensitivity with the result that the brake pressure can be much more accurately regulated.

The reset device is also of critical importance because at low speeds, e.g., below 10 m.p.h., the governor forces can become so diminished that the governor can behave entirely as if the vehicle were skidding, though not, thereby totally relieving the brake pressure. The reset device can act before this occurs, whereby normal control is regained.

A further aspect of my invention concerns the realization that a predetermined level of brake pressure, e.g.,

the minimum brake pressure at which skidding of the vehicle will occur on ice, generally between 100-150 p.s.i. for conventional automobiles, can be utilized to determine the setting for the brake pressure-sensitive reset device. Thereby, only at the time when skidding is no longer possible will control be taken away from the anti-skid unit to return the brake system to normal operation. This pressure-sensitive device, particularly when taking the form of a brake fluid pressure-sensitive switch, is useful with anti-skid systems generally, though of particular usefulness in the system described above.

Advanta[illegible] the brake pressure-sensitive switch operates an [illegible]ect ed system, preferably a solenoid, to restore norm[illegible]e operation, and it can be employed to bring t[illegible]d unit into readiness whenever the operator ap[illegible]es his brakes, as well as to stop the anti-skid un[illegible] from acting when skid conditions have ended.

Another aspect of the invention concerns the control of the governor. In normal operation of the brakes, the governor is prevented from influencing the brakes, but once activated, e.g., by the inertia-sensitive device, it is made continually to act, even when the vehicle wheel speeds up after the brakes have been initially relieved. A controlling or restraining member, preferably a solenoid, is employed to achieve this operation, with the controlling member held steadily from interfering with the governor until the reset device operates. In the case where a rotary weight mechanical governor is employed, a plunger can normally restrain the governor and a solenoid, initially under the control of the inertia-responsive device, can pull back the plunger to free the governor, whereupon the plunger can operate a switch to maintain the solenoid energized, and at the end of skid conditions the reset device can de-energize the solenoid to once again restrain the governor.

Another aspect of the invention is a fluid valving system that responds to the governor to increase and decrease brake pressure. Advantageously, the valve, discharging to a brake control conduit, is moved by the governor alternately between two positions to connect two passages for fluid at different pressures to the conduit. This causes the brake pressure to rapidly vary in a modulating or hunting manner, to keep the brake pressure at maximum values while preventing skidding. The mean about which the pressure modulates changes as the slipperiness of the road changes, due to the effect such road changes have upon the governor.

For use in a closed-circuit hydraulic system, the direct control of the brakes is achieved, according to the invention, by an isolation valve that continually seals the wheel brake from the control of the operator, and a variable volume chamber, e.g., piston and cylinder, for relieving and reapplying brake pressure to the wheel, the piston modulating about a mean determined by the modulating pressure conditions of the brake control conduit. Advantageously, the piston, at the position where it allows full pressure, can operate the isolation valve. Then the piston being spring loaded to that position can also perform a pressure-limiting function, limiting brake pressure to, e.g., 1,000 p.s.i., though the operator is capable of applying twice that much. In normal operation the brakes function as well at 1,000 p.s.i. as above, and by so limiting the brake pressure, less pressure is present to be relieved when a skid occurs, which enables faster action and allows the relief chamber to be smaller.

In other brake systems the brake control conduit can control a power unit, or in the case of air brakes, it can directly control the brake actuators associated with the wheels.

Furthermore, according to the invention, I have provided advantageous and improved constructions of the various operating components and sub-systems which can best be described with reference to the drawings.

Referring now to the drawings, and more particularly to FIGS. 1-6 thereof, a closed hydraulic circuit braking system employing my novel and improved anti-skid means in combination with a so-called nonpower-brake-will be described. The over-all system, generally designated as 10, comprises brake pedal 12, master cylinder 14, brake-relieving device 16, hydraulic pressure switch 18, relief actuator means 20, conduit valve means 22, skid-sensing and control means 24, and anti-skid unit drive means 26.

Referring to FIGS. 1 and 2, the brake pedal 12 is of conventional construction and comprises bell crank 28 pivotally [illegible]nected to a suitable support on the vehicle (not s[illegible] at 30 whereupon depression of pedal 12 causes bell crank 28 to rotate in a counterclockwise direction around pivot point 30, thus actuating connecting rod 32 pivotally connected to bell crank 28, as at 34. Rod 32 carries at its other extremity a piston 36 which functions to pump oil or other suitable fluid from master cylinder 14, all in a well-known manner. Specifically, master cylinder 14 has an upper reservoir 38 which communicates with lower chamber 40 by means of opening 42 whereupon a full supply of oil is always present in the chamber 40 to be pumped by actuation of pedal 12 and piston 36.

Fluid pumped from master cylinder 14 passes through a conduit 44 connected at one of its extremities by suitable couplings 45, 48 to chamber 40 of master cylinder 14 and at its other extremity by a suitable coupling 50 to the brake-relieving device generally designated at 16, which incorporates an isolation valve. More specifically, coupling 50 is connected to a second coupling 52 which is threadedly attached to block 54, said block having a bore 56 extending longitudinally therethrough into communication with aligned longitudinally extending bores 58 and 60 in couplings 52 and 50, respectively.

As will be seen most clearly in FIGS. 2 and 3, coupling 52 is provided with an internal shoulder for receiving positioning spring 62 which positions isolation valve ball 64 adjacent valve seat 66, tending to close with slight force the bore or passageway 56. During normal operation of the vehicle, ball 64 is maintained in its unseated position, as illustrated in FIGS. 2 and 3, against the action of spring 62, by means of spring-loaded actuator piston 68 having a reduced extension 70 in engagement with ball 64. Suitable O-rings or the like 72 are provided to insure a tight sliding fit between piston 68 and bore 56 whereby said piston effectively blocks further flow of fluid through bore 56.

Block 54 is provided with a pair of oppositely disposed passageways 74 and 76, it being noted that said passageways communicate with bore 56 just behind valve seat 66, whereupon when ball 64 is in its closed position in engagement with valve seat 66, fluid in conduit 44 from the master cylinder will be blocked before reaching the passageway 74 and 76. Passageway 74 communicates with hydraulic pressure reset switch 18 for reasons hereinafter to be described, while passageway 76 communicates by means of suitable couplings with conduit 78, which in turn communicates with the hydraulic brake lines 80 leading to the wheel hydraulic actuator cylinders (not shown) of the vehicle to actuate the brakes in well-known fashion. Thus it will be seen that with the parts in the positions illustrated in FIGS. 2 and 3, and particularly with piston 68 in the position illustrated, brake pedal 12 controls actuation of the vehicle brakes through a direct line. This is the normal operating condition of the vehicle.

The actuator for the relieving device 16, generally designated at 20 (FIGS. 1, 2) controls the movement of piston 68 and hence ball 64. This actuator comprises a housing or cylinder 82 having a large piston 84 slidably mounted therein, said piston having suitable seals 86 at its outer periphery to insure that the piston 84 will make a tight sliding fit within cylinder 82 whereby no leakage will exist around the piston. Piston 84 is threadedly connected, as at 88, to the aforementioned small hydraulic piston 68, whereupon movement of large piston

84 causes corresponding movement of piston 68. Housing 82 is secured to block 54 by any suitable means, such as screws 90, and the adjacent end wall 92 of the housing 82 is in free communication with atmosphere, as by ports 94. Suitable air filters, such as shown at 95 in FIG. 1, may be associated with the ports 94. It will thus be seen that the chamber 96 located on one side of large piston 84 is always at atmospheric pressure, while chamber 98, located on the opposite side of piston 84, may either be at atmospheric pressure or it may be vacuum charged, as will hereinafter be described in more detail. When the chambers 96 and 98 are both at atmospheric pressure, spring 100 resiliently urges piston 84 to the position illustrated in FIG. 2, it being seen that the piston 84 has moved to the right as far as it can go. In this position, the piston 68 has also been axially moved to the position illustrated, in which position extension (FIG. 3) has engaged ball 64 to unseat it from valve seat 66. Thus, when chambers 96 and 98 are both at atmospheric pressure, the vehicle brakes are under the complete control of the brake pedal 12. However, when chamber 98 is vacuum charged, the pressure differential between chambers 96 and 98 is sufficient to overcome spring 100, whereupon piston 84 will move to the left, viewing FIG. 2, thus causing corresponding movement to the left of piston 68. This in turn enables ball 64 to seat against valve seat 66, by means of spring 62, thus blocking the bore 56 and preventing actuation of the brakes by means of brake pedal 12. As pistons 84 and 68 move further to the left due to the presence of a vacuum or even a partial vacuum in chamber 98, hydraulic pressure in lines 78 and 80 and in the spring-loaded brake actuators associated with the vehicle wheels will be relieved, due to the fact that the fluid is now free to back up into that portion of bore 56 that has been vacated by movement of piston 68 to the left. It will therefore be seen that by introducing a vacuum or partial vacuum to chamber 98, fluid pressure in lines 78 and 80 will automatically be relieved, thereby relieving the brakes. By the same token, as soon as atmosphere is again introduced to chamber 98 to equalize the pressure in chamber 96, the brakes can once again be fully actuated by the pedal 12. It will be apparent that the amount of vacuum introduced to chamber 98 will determine the amount of reduction in brake pressure. More specifically, the greater the vacuum introduced to chamber 98, the greater will be the movement of piston 84 to the left (viewing FIG. 2) and at the same time there will be greater movement of piston 68 to the left, thus vacating more of bore 56 and providing greater relief of the hydraulic brake pressure. Similarly, given a certain amount of relief due to a certain amount of vacuum, then decrease in the amount of vacuum will increase the hydraulic brake pressure due to movement of the pistons to the right. The ability of my system to provide varying degrees of relief and restoration of brake pressure while the brake pedal remains isolated from the system is an important and advantageous feature.

It will be noted that during normal operation of the system, as heretofore described, i.e., where atmospheric pressure exists in both chambers 96 and 98, the piston 68 will function as a pressure regulator in the system. More specifically, since the piston is resiliently held in the position illustrated in FIG. 2 by means of spring 100, it will back off when excess pressure is applied by foot pedal 12. In practice it has been found that in a conventional automobile pressure above 1,000 p.s.i. or at least above 1,500 p.s.i., depending on the type of vehicle, is unneeded for normal brake operation and hence springs 62 and 100 are preferably calibrated so as to enable ball 64 to seat when pressure in excess of that needed is introduced, thus preventing introduction of excessive pressure to the brakes. It will be obvious that the springs may be calibrated so as to provide a limit at any selected pressure.

The means for introducing either atmospheric pressure or vacuum to chamber 98 and for modulating the degree of vacuum therein will now be described. Referring to FIG. 2, it will be seen that cylinder housing 82 is provided with an opening 102 in communication with chamber 98, which opening communicates with a brake control conduit 104 that in turn connects with the conduit valve, generally designated at 22. More specifically, there is provided a housing 106 having an upper portion 108 which houses the skid-sensing and control means 24, and a lower portion 110 which houses the valve 22. The lower housing 110 comprises an inner chamber 112 having a partition 114 defining a vacuum passage 116 and an atmosphere passage 118. More specifically, vacuum passage 116 is in communication, by means of passage 119, with the intake manifold 120 of the vehicle. Thus, suction is constantly being applied to the passage 116. The passage 118, on the other hand, is in communication with atmosphere through port 122, to which a suitable air filter 124 may be attached, if desired. A rotary valve 126 is mounted for rotation in housing 106, it being noted that valve 126 is an elongated rod that extends vertically through upper partition 128, the aforesaid partition 114, and is journaled in bottom wall 130, the partition and wall being integral parts of a unique molded housing of anti-friction plastic. The valve 126 is provided with a pair of passageways or bores 132 and 134 extending diametrically therethrough, said bores being vertically spaced from each other, and having elongated cross-sections in the axial direction, longer than the diameter of the rod. Bore 132 is associated with passage 116 and bore 134 with passage 118. The bores are angularly disposed with respect to each other, as shown most clearly in FIG. 6. Thus, with the rotary valve 126 in the position illustrated in FIGS. 2 and 6, bore 134 is positioned so as to allow free passage of atmosphere from passages 122 and 118, bore 134 and chamber 112 to brake control conduit 104, whereby atmosphere will flow into chamber 98. At the same time, due to the angular displacement of bore 132, passage 116 will be closed by valve 126, thus preventing suction from manifold 120 and passage 119 from coming into contact with chamber 112 and brake control conduit 104. It will be understood that the only access from passage 116 and 118 to chamber 112 is through valve 126 and more specifically, the bores 132, 134 thereof. Should valve 126 be rotated by means hereinafter to be described, it will be seen that communication between passage 118 and chamber 112 will eventually be blocked, thus interrupting the flow of atmosphere to brake control conduit 104, while at the same time passage 116 will be in communication with chamber 112, whereupon a suction will be applied through brake control conduit 104 to chamber 98.

It is important to note that use of a rotary valve such as valve 126 is extremely advantageous where passage of a vacuum and atmospheric pressure, or more generally two different pressures, is being controlled due to the fact that the pressure differential offers a minimum of resistance to the turning movement of the small diameter valve. This is contrasted to a reciprocal or poppet type valve wherein movement of the valve is much more resisted by pressure differential.

As hereinbefore explained and described, introduction of a vacuum or partial vacuum to chamber 98 automatically tends to release brake pressure on the vehicle wheels. It has been found in practice that an angular displacement between bores 132 and 134 of approximately 37.5° gives best results in the operation of my system, and it will be understood that rotary valve 126 may not always be positioned so as to completely open or close the bores 132 and 134. Thus, different degrees of movement of valve 126 allow different brake fluid pressures to be exerted on the vehicle wheels, thus providing the proper braking pressures for different road conditions. In instances where the pressure on the brake actuator is the same at the beginning of a skid, (lesser initial pressure requires less

relieving), the amount of relief of the brake pressure depends on the amount of travel of piston 84 to the left in chamber 98 which in turn is dependent on the amount of vacuum introduced to the chamber. Thus, where the vehicle is braked and skids on a dry surface which requires a greater brake pressure to create a skid condition, lesser movement of piston 84 to the left, and hence less vacuum, will be required to relieve the brake pressure sufficiently to prevent the vehicle wheels from locking, this pressure being referred to as "release" pressure. Conversely, where the vehicle is braked or a slippery surface such as ice, a lesser brake pressure is required to create a skid condition. Hence the valve 126 must open suction passage 116 to the brake control conduit 104 for a longer initial period to provide greater vacuum to chamber 98 in order to move piston 84 sufficiently to the left so that sufficient relief of the brake pressure will be effected to prevent the wheels from locking.

In accordance with the invention after release pressure is attained, with corresponding movement of the piston then the governor causes the degree of vacuum in conduit 104 to modulate, which modulates the brake pressure through a narrow range of intermediate pressures above and below the release pressure, without ever restoring the normal, higher brake pressure until the skid condition ends.

To thus operate the valve, the upper extremity of valve 126 (FIG. 2) is provided with a reduced portion 136 that extends upwardly through partition 128, said portion 136 having a pin 138 extending therethrough to which is attached a circular projection 140, it being seen that as a longitudinal thrust is exerted against projection 140, the valve 126 will be caused to rotate.

The means for controlling movement of valve 126, namely, the skid-sensing and governor control means and the drive means therefor, will now be described. Referring to FIG. 4, skid-sensing and governor control means is designated generally at 24, and the drive means therefor is designated generally at 26. Referring first to the latter, the vehicle drive shaft is shown at 142, it being understood that the drive shaft is directly connected to the wheels of the vehicle (not shown) whereupon when the vehicle wheels lock, the drive shaft also locks. The drive shaft drivingly engages speedometer gear 144, which in turn is fixed to shaft 146, which in turn is drivingly connected to shaft 148 by means of a coupling shown at 150, said shaft 148 connecting with speedometer cable 152. Gear box 154 through which shaft 148 passes and in which the shaft 148 is suitably journaled, houses drivingly engaged bevel gears 156, 158, it being understood that bevel gear 156 is keyed to shaft 148 and bevel gear 158 is keyed to shaft 160, whereupon rotation of shaft 148, transmitted thereto by the vehicle drive shaft 142, is transmitted to shaft 160. Shaft 160, in turn, is drivingly interconnected, as at 162, to flexible shaft 164, said flexible shaft being connected, as at 166, to the unit 24, and said flexible shaft being drivingly interconnected with a shaft 168 which has fixed thereto a drive gear 170. It will therefore be seen that the gear 170 is drivingly interconnected with the vehicle drive shaft 142, which in turn is connected to the vehicle wheels. Thus, rotation of the vehicle wheels will cause corresponding rotation of gear 170, and, conversely, sudden slowing of the vehicle wheels, such as would occur where the vehicle approaches a skid condition by the operator applying more brake pressure than is required for a particular road surface, will cause not only the main drive shaft 142 to suddenly slow, but also drive gear 170. Gear box 154, and adaptor 172, by means of which the gear box is connected to the engine housing 174, comprise an assembly which may be easily and readily adapted to a conventional vehicle for driving the sealing and governor control means now to be described.

The unit 24 (FIGS. 1, 4) comprises a molded plastic housing 176 having a pair of spaced partitions 178, 180 therein defining chambers 182, 184 and 186. A solenoid 188 is secured to housing 176 adjacent chamber 182 by any suitable means, the solenoid 188 having a plunger 190 with a reduced extension 192 resiliently urged into chamber 182 by means of spring 194. When solenoid 188 is not energized, spring 194 bears against flange 196 affixed to plunger 190 to resiliently urge extension 192 into chamber 182. Suitable stop means 198 mounted on the interior of chamber 182 engaged flange 196 to limit resilient axial movement of plunger 190 and extension 192 at a predetermined point, for reasons hereinafter to be explained more fully. Journaled in the partitions 178 and 180 are a pair of aligned shafts 200 and 202, respectively. Shaft 202 is provided with an enlarged portion 204 having an internal bore for slidably receiving in telescoping relation a reduced extension 206 of shaft 200. Attached to shafts 200 and 202 is a governor assembly comprising a plurality of weights 208, each pivotally connected to a pair of links 210, which links in turn are pivotally connected to arms 212. The arms 212 located on one side of the weights 208 are affixed to shaft 200 as at 214, while the arms on the other side of the weights are affixed to shaft 202 as at 216. As hereinbefore pointed out, shaft 202 is journaled for rotation in partition 180 but is incapable of longitudinal movement. Shaft 200, on the other hand, is drivingly engaged with shaft 202 so as to rotate therewith, but said shaft 200 is axially slidable with respect to the said shaft 202. Since the governor assembly is affixed on one side to shaft 202 and on the other side to shaft 200, it will be seen that the speed of rotation of the shafts will determine their longitudinal position relative to each other. Spring 218 resiliently urges the governor assembly to its collapsed position wherein shafts 200 and 202 are axially extended with respect to each other, but as the rotational speed of shafts 200 and 202 increases, centrifugal force urges the weights 208 outwardly, thus moving shaft 200 toward the right, viewing FIG. 4 against the action of spring 218.

Fixed to shaft 200 at the extremity thereof located in chamber 182 is a spool member 220, which spool member is in engagement with projection 140 carried by the reduced upper extension 136 of rotary valve 126. Thus, axial movement of shaft 200 causes like movement of the spool 220 which in turn carries therewith projection 140 captured between the opposed walls of the spool to thereby impart rotary movement to valve 126.

The means for imparting rotational movement to shafts 200 and 202 and the governor assembly associated therewith comprises a cam gear 222 (FIGS. 4, 5 and 5A), which cam gear is part of the drive train, in driven engagement with gear 170. The cam gear 222 is mounted on shaft 202 and is located in chamber 186. The cam gear is slidably mounted on shaft 202 and is in driving engagement therewith by means now to be described. A recess 224 having inclined or helical surface 226 is provided at the end of the cam gear adjacent partition 180, and a projection or pin 228, carried by shaft 202, extends into said recess in engagement with the cam surface 226. At either end of the recess is a stop surface, at the deeper end stop surface 227, adapted to engage pin 228 for accelerating the governor and at the shallower end stop surface 225 for engaging pin 228 to decelerate the governor. A leaf spring 230 mounted to the housing 176 by any suitable means engages a button 232 which is press fitted into the central bore of cam gear 222 to normally urge the latter axially toward partition 180, holding pin 228 against accelerating stop surface 227. However, when rotation of drive gear 170 and cam gear 222 is suddenly slowed, as by sudden slowing of the vehicle wheels upon the presence of a skid condition, the rotational inertia of shaft 202, especially due to the governor weights 208, causes shaft 202 to turn ahead of gear 222, causing pin 228 to leave stop surface 227 and move to stop surface 225. This slides gear 222 on shaft

202 in a direction away from partition 180 to activate governor control.

It is important to note that solenoid spring 194 is stronger than governor spring 218, whereupon when solenoid 188 is not energized, plunger extension 192 bears against spool 220 to restrain same in the position illustrated in FIG. 4 wherein the rotary valve 126 is in the position illustrated in FIG. 2, namely, the position where atmosphere is free to pass through the valve to conduit 104 and then to chamber 98 and wherein at the same time suction from the intake manifold is blocked from communication with conduit 104. This is the normal operating condition of the system wherein the balanced pressure on opposite sides of actuator piston 84 enables spring 100 to position piston 84 and piston 63 as illustrated in FIG. 2. In this position, ball valve 64 is unseated to allow fluid pressure to pass directly to the brake line whereupon operation of the brakes is under direct control of brake pedal 12. This condition will remain so long as solenoid 188 (FIG. 4) is not energized. Energization of solenoid 188 is accomplished in the following manner. As previously explained, sudden slowing of gear 170 due to sudden slowing of the vehicle wheels as a result of a skid condition causes cam gear 222 (FIGS. 4, 5, 5A) to move axially on shaft 202 away from partition 180 against the action of spring 230. This movement is sufficient to cause actuation of a switch 234 suitably mounted in chamber 186. At the same time, the initial movement of the brake pedal and the resultant increase in fluid pressure in the system; causes hydraulic switch 18 (FIG. 2) to become closed. Hydraulic switch 18 is fully disclosed and described in copending U.S. patent application Ser. No. 203,075, filed June 18, 1962. Basically, switch 18 comprises piston 236 which has connected thereto at its upper end a metallic contact cup 238. Cup 238 is adapted to bridge contacts 240 and 242 but is normally maintained spaced therefrom by spring 244. An adjusting screw 246 is provided by means of which the tension of spring 244 may be varied, thus enabling the switch 18 to be preadjusted whereby a predetermined fluid pressure, e.g. 120–125 p.s.i., will cause piston 236 to move against the action of spring 244 sufficiently to bridge the contacts 240, 242. Actuation of switch 18 in the aforedescribed manner enables current to flow from battery 248 through contact 242, contact cup 238, and contact 240 to line 250 to the solenoid coil positive terminal 252 (FIG. 4). The voltage is applied through the solenoid coil 254 and then through line 256 to terminal 258 on switch 234. When switch 234 is open, the circuit to ground is not complete and solenoid 188 will not be energized. However, upon actuation switch 234 in the manner aforedescribed, the circuit to ground is completed through terminal 260, line 262 and ground connection 264. It is important to note that the circuit is such that although it is essential for hydraulic pressure switch 18 to be closed in order for solenoid 188 to be energized, closing of switch 18 alone is not sufficient, but rather switch 234 must also be actuated in order to complete the circuit. This is important because under normal operating conditions of the system sufficient pressure will normally be introduced to switch 18 by the foot pedal to actuate same, but it is essential that solenoid 188 not be energized under such normal conditions.

Energization of solenoid 188 accomplished by rightward movement of the cam gear (FIG. 5) closing switch 234, causes plunger 190, 192 to be retracted against the action of spring 194, thus freeing the shaft 200 and spool 220 for axial movement under the influence of the governor assembly. Therefore, the cam gear operating in response to the inertia of the governor and the solenoid operated plunger together can be regarded as an inertia-responsive means for initiating the action of the governor.

As soon as the solenoid plunger has been retracted, due to the sudden slowing at the commencement of a skid condition, the governor assembly takes over control and, as it is stopped by stop surface 225 of the cam gear, spring 218 will cause the governor assembly to commence to collapse, thus moving shaft 200 and spool 220 axially to the left, when viewing FIG. 4. This movement of spool 220 causes corresponding rotary movement of valve 126, thus shutting off the passage of atmosphere to brake control conduit 104 and simultaneously effecting communication between vacuum passage 119 and conduit 104. This results in suction being introduced to chamber 98 which in turn overcomes the action of spring 100 to cause pistons 84 and 63 to move to the left, viewing FIG. 2. This movement of piston 63 enables ball valve 64 to close due to the action of spring 62, thus preventing the introduction of further fluid pressure to the system via brake pedal 12. Further leftward movement of piston 63 enables the fluid in lines 73 and 80 to back up, thereby relieving the pressure until the vehicle brakes begin turning, the duration that the suction is applied, and hence the degree of vacuum and the amount of pressure relief depending on the slipperiness of the road. As soon as the brake release pressure is achieved, the vehicle wheels commence to rotate, whereby drive shaft 142 through the aforedescribed gearing once again causes gears 170 and 222 to rotate, it being apparent that this renewed rotation of shafts 200 and 202 will cause the governor assembly to once again turn, and due to the centrifugal force created, to retract shaft 200 and spool 220 to once again rotate valve 126 to the positions illustrated in FIGS. 2 and 4, where once again atmosphere is connected to chamber 98, thus enabling brake pressure to be restored.

But, it is very important for my present invention that the solenoid is not de-energized and the governor returned to the restraint of the inertia-responsive device. Rather, the governor remains free to act. It leaves the valve 126 open to atmosphere 118 for an instant until the degree of vacuum is lessened enough to cause slight rightward movement of the piston 84 to cause the brake pressure to slightly exceed release pressure and a skid again commences. Again the governor is slowed by the drive train and again it opens the valve 126 to suction 116, and this cycle repeats again and again in an extremely rapid hunting or modulating action with the gas pressure in the conduit 104 never reaching either atmospheric or engine vacuum level, but rather modulates at intermediate levels. In dependent fashion, the hydraulic brake pressure modulates narrowly about the release pressure as a mean, without returning to normal pressure. When the skid condition ends, the governor opens the valve continually to atmosphere and pistons 84 and 85 move fully to the right to once again unseat isolation valve 64 and restore normal brake operation. The faster the vehicle travels, the faster does this action occur, but the action is sufficiently fast down to speeds on the order of 10 m.p.h. and even lower, to rapidly pump and release the brakes. It is another very important aspect of the invention that the governor relinquishes control of the brake pressure before the vehicle stops.

In order to effectively carry out this modulating action, I have found it to be critical, in the preferred construction, that the solenoid 188 remain energized or that the plunger in some other way remains retracted until the vehicle slows to a predetermined slow speed, e.g. 5 or 10 m.p.h., depending on the vehicle. Since closing of switch 234 to complete the circuit to ground that energizes solenoid 188 is only momentary, holding means are provided for maintaining the solenoid energized until hydraulic pressure switch 18 opens, it being understood that switch 18 is preset to open only when fluid pressure in the system falls below the minimum value possible to achieve a skid condition, e.g. set to 85–90 p.s.i. produced by the governor at 5 or 10 m.p.h.

The aforesaid holding means for the solenoid comprise a terminal 266 having a connection to ground as at 268, which terminal is engaged by plunger 190 when the latter is retracted in order to complete the circuit to ground independent of switch 234. Thus, although closing of switches 18 and 234 is essential in order to energize solenoid 188, once the solenoid has been energized and its plunger retracted, switch 234 may be opened without de-energizing the solenoid, and the solenoid will remain energized until switch 18 opens.

As hereinbefore stated, it will be obvious that if switch 18 were not utilized, then there would be no means present for insuring de-energization of the solenoid once the condition has been reached where a skid is no longer possible, it being obvious that if the solenoid were still energized at such low speeds, the vehicle would be without effective brakes, since the governor assembly would not exert sufficient centrifugal force to overcome the action of spring 218 and move shaft 200 and spool 220 to the rightward position (FIG. 4) wherein valve 126 enables the brakes to be controlled by pedal 12.

It is possible for the reset device to take some other form, such as a pendulum device or a vehicle inertia memory system that can detect the slow speed of the vehicle, but brake pressure-sensitive devices, and particularly hydraulic-electric switches have important advantages in the system, particularly since they can respond whenever the skid condition has passed, regardless of the speed of the car, to enable normal operation, while offering low speed protection against the governor's cutting off the brakes completely. The adjustment of the pressure level to which the device responds can be adjusted according to the wishes of the operator and the type of road conditions to be expected. But, in fact, a single setting, below that at which skidding on ice occurs, which is determinable for any given vehicle, is advantageously employed to give anti-skid protection under virtually all conditions.

Any tendency for the unit 24 to erroneously sense a skid condition and actuate switch 234 in response to a rapid deceleration is overcome by use of properly calibrated weights 206 and a properly tensioned spring 230. In addition, I have found that an erroneous sensing is most likely to occur where there is a rapid acceleration of the drive shaft quickly followed by a rapid deceleration. The frictional brake or drag that is resiliently exerted on spool 220 by plunger 190 helps to overcome this problem since this action tends to reduce the torque applied to shafts 200 and 202 by the governor assembly. It has been found, however, that at high speeds the frictional engagement between plunger 190 and spool 220 results in undue wear of the parts. Therefore, plunger 190 is assisted by spring 230, and the plunger is disengaged at high speeds. To this end, stop means 198 engages flange 196 so as to limit the outward movement of plunger 190 whereupon no engagement exists between plunger 190 and spool 220 at relatively high speeds. I have realized that the crucial problem of rapid acceleration followed by rapid deceleration occurs at low speeds in the low gear range with conventional automobiles, and I have, therefore, found that it is not necessary to have plunger 190 exert a frictional drag on spool 220 and shaft 200 at relatively high speeds.

Although the operation of my invention is thought to be clear from the foregoing description, a brief resumé of the operation will now be given. Under normal operating conditions, solenoid 188 is de-energized whereby the spring-loaded solenoid plunger maintains the skid-sensing and control unit inoperative. The de-energized solenoid further insures that rotary valve 126 is in the position wherein atmosphere is introduced to chamber 98. Since chamber 98 is also open to atmosphere, it follows that there is a pressure balance on each side of large piston 84 thus enabling spring 100 to take over and move the piston 84 to the position illustrated in FIG. 2. In this position, piston 63, which is connected to piston 84 and moves therewith, engages ball valve 64 so as to unseat same against the action of spring 62. with ball valve 64 so opened, fluid pumped from master cylinder 14 by brake pedal 12 is free to pass directly to hydraulic lines 78 and 80 to actuate the vehicle brakes. The pressure-sensitive switch is adapted to be closed by a low level of brake pressure, e.g. 120–125 p.s.i., placing the anti-skid unit into readiness for operation.

Upon encountering a skid condition, i.e., a condition wherein the vehicle operator applies more brake pressure than is required for a particular road surface, the vehicle wheels slow suddenly. This causes corresponding deceleration of drive shaft 142, and, through suitable gearing, drive gear 170 and cam gear 222 also slow abruptly. The inertia of the governor assembly causes the gear 222 to move longitudinally by the cam surface aforedescribed, said longitudinal movement of gear 222 causing switch 234 to be closed which results in energization of solenoid 188 and the retraction of solenoid plunger 190. The second switch or terminal 266 is engaged upon retraction of the solenoid plunger to maintain the solenoid energized until switch 18 opens, even though switch 234 has now opened. Switch 18 is preset, e.g. at 85–90 p.s.i. to be opened at a brake pressure below that at which skids can occur, e.g. 100–150 p.s.i. for ice, whereupon solenoid 188 will remain energized until possibility of skidding no longer exists, and the solenoid is then de-energized to restore normal foot brake action before the governor has a chance ever to totally relieve the brakes.

Regarding the operation of the governor, once the solenoid 188 has been energized, the governor control unit takes over. More specifically, when a skid condition is approached, the sudden slowing of the vehicle wheels causes the governor assembly to collapse under the action of spring 218, thus moving shaft 200 and spool 220 to rotate valve 126, this movement now being permitted due to retraction of plunger 190. Rotation of valve 126 blocks the flow of atmosphere to chamber 98 and simultaneously allows the introduction of a vacuum to said chamber. The introduction of vacuum overcomes spring 100, thus resulting in movement of pistons 84 and 63 to the left, viewing FIG. 2. This movement closes isolation valve 64, thus preventing the application of further fluid pressure by brake pedal 12. As the valve remains in this position, the degree of vacuum increases, causing further leftward movement of piston 63 (FIG. 3), so that pressure in lines 78 and 80 and the brakes are relieved. A point is reached when the brakes are relieved sufficiently to allow the wheels to speed up from their slowed condition, the degree of relief required depending on the degree of slipperiness of the road. Then shafts 200 and 202 speed up once again, whereby the governor assembly retracts shaft 200 and spool 220 to once again rotate valve 126 to the position where atmosphere is introduced to chamber 98 and the piston 63 moves to the right and increases the brake pressure, until a part is reached where the wheels slow again and the cycle repeats. This action continues in a hunting manner so long as the vehicle is in an incipient skid, thus insuring that maximum brake pressure is continually applied to the wheel brakes short of pressure that would cause locking of the wheels. Once the brake pedal is released or the brake pressure in the system has dropped below the minimum possible value to create a skid, hydraulic pressure switch 18 breaks the circuit to solenoid 188, whereupon the latter becomes de-energized and the brakes once again become under complete control of the operator.

It is important to note that although loss of vacuum in the vehicle system will render the anti-skid means ineffective, said loss of vacuum will in no way affect normal operation of the vehicle brakes. Also, it is important to note that this system can be used with foot pedal controlled brakes of both the powered and nonpowered types.

While a spring-loaded fly ball type of governor has been illustrated, and is presently preferred, the broad concept

of the invention include the use of other governors, all rotary weight governors, such as gravity loaded fly ball, centrifugal shaft and inertial shaft governors, as well as governors that operate through electrical action.

The concepts are likewise applicable to other brake systems including air brakes in which valve 126 could directly modulate air brake pressure. Likewise it is possible to separate the governor from the valve, let the governor actuate a switch and let the valve be actuated by a solenoid responsive to the condition of the switch.

While use with engine vacuum is very efficient and inexpensive, and unlike previous systems I have proposed, does not cause the engine to run roughly due to loss of vacuum, the invention is applicable to other sources of pressure differential, incompressible fluid as well as gaseous fluid and numerous aspects of the invention are useful with nonfluid types of brake systems.

Referring now to FIGS. 7 and 8, a slightly different form of my invention is disclosed in which the relieving device is incorporated as an integral part of a power brake unit. The system generally comprises a brake pedal 309, a master cylinder 302, an accumulator 304, a safety valve 306, a hydraulic pressure switch 308, a power brake control unit 310, and an anti-skid unit 312.

Actuation of brake pedal 309 operates piston 314 to pump fluid from cylinder 316 to conduit 315. Master cylinder 302 further comprises a fluid reservoir 320 communicating with cylinder 316 by means of passageway 322. A conduit 324 extends from reservoir 320, the function of which will hereinafter be described. Accumulator 304 is in communication with conduit 315 and comprises a spring loaded piston 326 and acts as a cushion and shock absorber for the fluid system where variations in fluid pressure occur.

During normal operation of the vehicle with which the power brake unit is associated, and with the brakes in their non-actuated position, the power brake unit 310 is in the position illustrated in FIG. 8. In this position, chambers 328 and 330 of cylinder 332 are both under vacuum, thereby equalizing the pressure on opposite sides of piston 334. This enables spring 336 to urge the piston 334 to the leftward position illustrated, said piston 334 being connected by means of rod 338 to a brake actuating piston 340. Piston 340 is slidably mounted in cylinder 342, said cylinder having an outlet conduit 344 leading to safety valve 306. Conduit 344 is also in communication with hydraulic pressure switch 308.

A supply of hydraulic fluid is located in cylinder 342 whereupon movement of piston 334 against the action of spring 336, and corresponding movement of piston 340 will force fluid from cylinder 342 under pressure through conduit 344 and with valve 306 in the position illustrated in FIG. 8 to conduit 346 and then to hydraulic lines 348 which lead to the wheel cylinders (not shown) of the vehicle's braking system. Thus it will be seen that movement of pistons 334 and 340 to the right, viewing FIG. 6, will cause operation of the vehicle brakes.

As hereinbefore indicated, under normal operating conditions of the vehicle, and with the brakes not applied, pistons 334 and 340 are in the position illustrated in FIG. 8 due to the fact that a vacuum exists in both chambers 328 and 330. More specifically, suction from the intake manifold 360 of the vehicle passes through conduit 352 into chamber 330 and then through passageway 354, through openings 356 in diaphragm 358 to conduit 360, through rotary valve 362 of the anti-skid unit 312, through conduit 364 and then to chamber 328. The control unit generally shown at 366 which forms a part of power brake unit 310 is of conventional form and comprises the aforesaid diaphragm 358, hydraulically movable by means of piston 368 to a closed position wherein the diaphragm engages the fixed seat 370 to block communication between passage 354 and conduit 360. The piston 368 also controls movement of poppet valve 372 normally maintained closed by spring 374. Spring 376 normally urges diaphragm 358 to its open or unseated position, as illustrated in FIG. 8. An atmosphere intake is located at 378, and introduction of atmosphere into the system is controlled by valve 372 in a manner now to be described.

As above indicated, with the parts in the positions illustrated in FIG. 8, the brakes of the vehicle are non-operative due to the fact that suction is free to circulate through conduit 352 to chamber 330 and then through control unit 366 to conduit 360, through rotary valve 362 to conduit 364 and then to chamber 328. With vacuum on both sides of piston 334, spring 336 maintains the piston 334 and the connected brake actuating piston 340 in brake release position. Upon actuation of brake pedal 309 by the operator, pressurized fluid passes through conduit 315 to conduit 350. With the safety valve 306 in the position illustrated, flow of the fluid into the valve is blocked, and hence the brake fluid is forced to flow through conduit 350 into the brake control unit 366. None of the fluid from the pedal is allowed to reach the brake actuators at 348.

Introduction of the pressurized fluid to the brake control unit 366 causes actuation of piston 368 to move diaphragm 358 to its closed or seated position and to simultaneously open poppet valve 372 in a conventional manner. This movement blocks communication between chamber 330 and conduit 360 and simultaneously allows air to enter through inlet 378 through the now open valve 372 to conduit 360. The atmosphere circulates around conduit 360 through valve 362 to conduit 364 and then the chamber 328. The introduction of atmospheric pressure to chamber 328 creates a pressure differential which overcomes spring 336 and moves piston 334 and piston 340 to brake actuating position. More specifically, the movement of piston 340 in cylinder 342 forces the brake fluid through conduit 344 and safety valve 306 to conduit 346 and hydraulic brake lines 348 to actuate the vehicle brakes in known fashion.

In order to insure that there is always an adequate supply of brake fluid in cylinder 342, conduit 324 connects fluid reservoir 320 to the cylinder 342, it being important to note that the inlet 382 to the cylinder 342 is located behind the forward portion of piston 340. Piston 340 is provided with a one-way valve 384 which enables fluid to pass through the piston from left to right, viewing FIG. 8, but which blocks flow of fluid through the piston from right to left. Thus, during operation of the piston 340, the suction created thereby in cylinder 342 sucks fluid through conduit 324 from reservoir 320 into cylinder 342 on the rear side of piston 340. During the return stroke of piston 340, the valve 384 allows fluid to pass through the piston, thereby insuring that there is always a full supply of fluid in the cylinder for brake actuation.

Since the structure and operation of hydraulic pressure switch 308 and anti-skid unit 312 are identical to that already described in connection with the first form of my invention, it is not thought that further description is necessary. It will be noted, however, that where the operator applies more brake pressure than is required for a particular road surface, the vehicle wheels will lock, whereby the same structure and operational sequence will cause rotary valve 362 to be moved to a position wherein conduit 360 becomes blocked and conduit 352A becomes open to allow suction from manifold 350 to be introduced through conduit 364 to chamber 328, whereupon spring 336 will once again take over to release the brakes. The same modulating action as aforedescribed will take place until hydraulic pressure switch 308 eventually renders the anti-skid inoperative.

In accordance with my invention it is crucial that the entire brake pressure be controlled by the anti-skid unit and that no fluid be conducted directly from the foot pedal to the wheels. Means are provided to insure that brake pressure will not be lost should there be a loss of suction. This is accomplished by safety valve 306. More specifically, conduit 352B is connected to the suction con-

duit 352 whereupon as proper suction exists in the system, piston 356 is maintained in the position illustrated in FIG. 8 against the action of spring 358. In this position, conduit 344 is in communication with conduit 346, but conduit 350 is blocked at its extremity adjacent the safety valve. Should suction be lost in the system, spring 358 will become effective to move piston 356 upwardly, thereby blocking the communication between conduits 344 and 346 and at the same time providing communication between conduit 350 and conduit 346. With the safety valve in this position, the brakes are directly controlled by operation of pedal 300 since the entire power unit is now bypassed.

A one-way check valve 390 is located in conduit 352 adjacent the intake manifold 350 since without same backfire of the vehicle engine would cause an explosive pressure to be introduced to the power unit, which would be undesirable for obvious reasons.

Referring once again to FIG. 4, it will be noted that should the vehicle move in reverse, the direction of rotation of drive shaft 142 will likewise be in reverse, thus causing reverse rotation of flexible shaft 164 and shaft 168. Reverse rotation of shaft 168 will, in turn, cause reverse rotation of drive gear 170 and cam gear 222, causing pin 228 to leave stop surface 227 and move toward stop surface 225, which, as previously described, causes gear 222 to slide on shaft 202 in a direction away from partition 180 to close switch 234 and thus activate the governor control and anti-skid means. Even though it is possible that very slow reverse movement of the vehicle might not be sufficient to cause movement of cam gear 222 to activate the governor control, the fact remains that there is always the possibility and even likelihood that reverse movement of the vehicle will activate the governor control even though no skid condition exists. Since there is no practical need for any anti-skid means when the vehicle is moving in reverse, and since the activation of these means when not needed might even result in the loss of some stopping distance, it has been found desirable to provide means for rendering the governor control and anti-skid ineffective and inoperative when the vehicle moves in reverse.

Referring to FIGS. 9 through 12, a somewhat, modified anti-skid unit is shown generally at 400. Unit 400 is basically the same as skid-sensing and control means 24 hereinbefore described except that a reverse cut-off switch has been incorporated in the unit to prevent energization of solenoid 188 when the vehicle drive shaft is in reverse and flexible shaft 164 correspondingly rotates in a reverse direction. The reverse cut-off switch, shown generally at 402 (FIG. 10) comprises a shaft 404 slidably coupled to shaft 164 as at 406, in the same manner that shaft 168 was drivingly engaged to flexible shaft 164. Shaft 404, which is constructed of any electrically conductive metal, is journalled in bore 408, said bore having an enlarged end portion 410, it being understood that the larger diameter of portion 410 results in the presence of an annular shoulder 412 at the extremity of bore 404. Slidably mounted in portion 410 is a contact button 414, said button being resiliently urged toward bore 404 by means of spring 416. The opposite end of spring 416 resiliently engages a contact cup 418 having a terminal 420 extending radially therefrom.

Adjacent its opposite extremity, shaft 404 has fixedly mounted thereon the afore-described drive gear 170, said gear meshing with cam gear 222, all in the manner hereinbefore described. A spring 422 engages washer 424 to normally urge shaft 404 to the left when viewing FIGS. 9 and 10, thus causing the end of shaft 404 to extend into enlarged portion 410 and, therefore, into engagement with contact button 414, as illustrated in FIG. 9. Loosely mounted on shaft 404 is a reverse switch cam 426 comprising a circular plate 428 having a spiral cam portion 430 mounted thereon, and further having an outwardly extending pin 432 located adjacent the end of cam track 430. A cam pin 434 extends radially from shaft 404 and is adapted to cooperate with the spiral cam track 430 to determine the axial position of shaft 404. Expressed differently, when shaft 404 is rotating in a counterclockwise direction when viewing FIG. 12, cam pin 434 will engage the stop pin 432 at the low point on the spiral track 430. With the parts in this position, shaft 404 will assume the position illustrated in FIG. 9 wherein the extremity of the shaft is in engagement with contact button 412, it being understood that spring 422 is somewhat stronger than spring 416, whereupon the extremity of shaft 404 will actually enter into enlarged portion 410. Upon clockwise rotation of shaft 404, pin 434 will ride up the spiral cam track 430 until it reaches the high end of said track and engages the opposite side of stop pin 432, as illustrated in FIG. 10. In this position, shaft 404 has been moved to the right (when viewing FIG. 10) against the action of spring 422 sufficiently for the end of said shaft to become completely withdrawn from enlarged portion 410. Since the annular shoulder 412 prevents the contact button 414 from moving out of enlarged portion 410, it follows that the end of shaft 404 becomes disengaged from contact button 414, as clearly illustrated in FIG. 10. Also mounted on shaft 404 is a washer 436 having a radially extending contact arm 438, the purpose of which will hereinafter be described. As will be seen most clearly in FIG. 10, the washer 436 is mounted on shaft 404 adjacent the flat rear surface of cam 426; it being understood that shaft 404 may freely rotate within washer 436. A cam thrust washer 440 comprising a peripheral ring portion 442 and bent marginal lugs 444 resiliently engages the opposite face 428 of cam 426, whereupon to resiliently urge cam 426 into engagement with washer 436 and at the same time to impart a sufficient frictional drag on cam 426 so that movement of shaft 404 will not impart corresponding rotary movement to the cam until pin 434 moves into engagement with stop pin 432. The thrust washer 440 has a radially extending spring arm 446 that is secured to the housing 176 by any suitable means, such as by having the end of the spring arm clamped between adjacent portions of the housing, as illustrated in FIGS. 9 and 10.

The operation of reverse switch 402 is as follows. Assuming that shaft 404 is being rotated in a counterclockwise direction, which for purposes of this description correspond to forward movement of the vehicle, the switch will, as aforestated, assume the position illustrated in FIG. 9 wherein shaft 404 is making engagement with contact button 412. With the parts in this position, and assuming that a skid condition commences to exist, causing cam gear 222 to close switch 234, current will flow through line 262 to terminal 420 through contact cup 418, spring 416, contact button 414, and then to shaft 404, which as previously described, is in engagement with contact button 414. Shaft 404 is in turn in electrical engagement with washer 436 and its radially extending terminal 438, both directly and through the engagement of pin 434, which in turn engages cam 426, which in turn bears against washer 436. Thus, the flow of current is through terminal arm 438 and then to line 262A, which connects to ground as at 264. This, along with line 256, completes the circuit to solenoid 188 in order to energize same and render the governor control effective, all in a manner hereinbefore described. On the other hand, should the vehicle move into reverse, thus resulting in clockwise rotation of shaft 404, cam 426 will cause the shaft 404 to retract and become disengaged from contact button 414, as illustrated in FIG. 10, thus interrupting the connection to ground and preventing energization of solenoid 188. It will thus be seen that whenever the vehicle and its drive shaft moves in reverse, the reverse cut-off switch 402 will automatically prevent the skid-sensing and control means and the governor control from coming into play. Thus, the vehicle brakes will always be under normal and direct control by the operator whenever the vehicle is moving in reverse.

While there is shown and described herein certain specific structure embodying the invention, it will be manifest to those skilled in the art that various modifications and rearrangements of the parts may be made without departing from the spirit and scope of the underlying inventive concept and that the same is not limited to the particular forms herein shown and described except insofar as indicated by the scope of the appended claims.

I claim:

1. A power brake system for motor vehicles comprising a plurality of hydraulic brake actuators, a master cylinder having a fluid supply therein, means for causing fluid to be pumped from said master cylinder, a power brake unit having a first cylinder and piston means dividing said cylinder into vacuum and pressure chambers, a source of vacuum connected to said vacuum chamber, a first conduit extending from said vacuum chamber to said pressure chamber, a control valve movable between a first position wherein suction from said vacuum side is free to communicate with said pressure chamber to evacuate same and a second position wherein atmospheric pressure is introduced to said conduit for entry to said pressure chamber, resilient means operative to bias said first piston to a first position when both chambers are evacuated, said first piston being movable to a second position against the action of said resilient means when atmospheric pressure is introduced to said pressure chamber, means for moving said control valve between its first and second positions in response to the pumping of fluid from said master cylinder, a second cylinder and piston, said second cylinder having a supply of fluid therein, a line connecting said second cylinder to said brake actuators whereby actuation of said second piston forces fluid under pressure to said brake actuators to operate same, means connecting said first and second pistons whereby movement of said first piston from its first to second position causes actuation of said second piston, a line connecting said second cylinder to said master cylinder, whereby the suction created by actuation of said second piston automatically draws fluid into said second cylinder from said master cylinder, and a one-way valve in said second piston for allowing fluid to pass therethrough as said second piston returns to its inoperative position, thereby insuring at all times an adequate supply of fluid in said second cylinder and anti-skid means for controlling fluid communication between said vacuum chamber and said pressure chamber upon initiation of a skid condition by the motor vehicle, said last named means being interposed between said control valve and said pressure chamber.

2. A power brake system for motor vehicles comprising a plurality of fluid brake actuators, a master cylinder having a fluid supply therein, means for displacing the fluid in said master cylinder, a brake control unit fluidly connected to said master cylinder and actuable by said displacing means, a first cylinder, a first piston mounted for endwise sliding movement in said first cylinder, means for fluidly connecting one side of said first piston to said brake actuators, means for actuating said first piston to move said first piston in a first direction to transmit fluid pressure to said fluid brake actuators, said last named means including said brake control unit and said fluid displacing means, a second cylinder, a second piston mounted in said second cylinder to divide said second cylinder into a first pressure chamber and a second pressure chamber, means for operatively connecting said first piston with said second piston, a first pressure source and a second pressure source, means for fluidly connecting said first pressure source with said first pressure chamber, said brake control unit including means for providing fluid communication between said second chamber of said second cylinder and said first pressure source when said brake control is not actuated, the difference in pressure between the two chambers causing said second piston to move in a direction to urge said first piston to move in the first direction, means for biasing said first piston in a second direction which is opposite to that of its first direction when both chambers of said second cylinder are in fluid communication with said first pressure source, means for closing fluid communication between said first pressure source and said second pressure chamber and placing said second pressure chamber in fluid communication with said second pressure source upon actuation of said brake control unit, and means interposed between said brake actuators and said first piston for providing direct fluid communication between said master cylinder and said fluid brake actuators upon a failure of said first pressure source to provide the pressure necessary to operate said second piston, whereby when said fluid displacing means is actuated fluid pressure is transmitted to said brake control unit thereby to actuate the brake control unit and close fluid communication between said first pressure source and said second chamber and open fluid communication between said second chamber and said second pressure source whereby said bias means is overcome and said first piston is moved in the first direction to apply fluid pressure to said brake actuators and whereby upon actuation of said means interposed between said brake actuators and said first piston the actuation of said fluid displacing means causes fluid pressure to be transmitted directly from said master cylinder to said brake actuators.

3. A power brake system as set forth in claim 2 wherein said means interposed between said brake actuators and said first piston includes a valve, means for actuating said valve to a first operating position to provide direct fluid communication between said fluid brake actuators and said first piston for actuating said valve to a second operating position cutting off the direct fluid communication between said fluid brake actuators and said first piston and opening direct fluid communication between said fluid brake actuators and said master cylinder.

4. A power brake system as set forth in claim 3 wherein said valve includes a housing formed with a piston receiving channel and a piston formed with spaced longitudinal grooves and mounted for endwise movement in said channel between a first position in said first valve operating position and a second position in said second valve operating position and said valve positioning means includes a spring adapted to bias said valve piston toward said second position, means for providing fluid communication between one end of said piston and said first pressure source and means for providing fluid communication between the other end of said piston and said second pressure source whereby in said first valve operating position said first and second pressures cooperate to overcome said spring bias to hold said valve piston in said first position but upon a failure of the first pressure said spring acts to move said valve piston to said second position.

5. A power brake system as set forth in claim 2 including anti-skid means for controlling fluid communication between said second pressure chamber and said first and second pressure source upon actuation of said fluid displacing means and said anti-skid means, said anti-skid means being interposed between said brake control unit and said second pressure chamber.

6. A power brake system as set forth in claim 5 wherein said anti-skid means includes means for sensing a predetermined deceleration of a wheel of the motor vehicle and means actuated by said sensing means for alternately removing fluid communication between said second pressure chamber and said second pressure source and opening fluid communication between said second pressure chamber and said first pressure source.

7. A power brake system as set forth in claim 6 including means for removing from the motor vehicle operator control of the amount of the fluid pressure applied to the brake actuators upon actuation of said said sensing means, whereby fluid pressure for said brake actuators is alternately applied and relieved skid control of the amount of fluid pressure applied to the brake actuators is removed from the operator.

8. A power brake system as set forth in claim 7 wherein said means interposed between said brake actuators and said first piston includes a valve, said valve including a housing formed with a piston receiving channel and a valve piston formed with a pair of spaced longitudinal grooves and mounted for endwise movement in said channel between a first position and a second position, means for moving said valve piston to the first position to provide fluid communication between said fluid brake actuators and said first piston and for moving said valve piston to the second position to cut off fluid communication between said fluid brake actuators and said first piston and open direct fluid communication between said fluid brake actuators and said master cylinder, said last named means including means for providing fluid communication between one end of said valve piston and said second pressure source and for providing fluid communication between the other end of said valve piston and said first pressure source and a spring adapted to bias said valve piston toward the second valve piston position, whereby in said first valve piston position said first and second pressures cooperate to overcome said spring bias to hold said valve piston in the first position but upon a failure of the first pressure said spring acts to move said valve piston to said second position.

9. A power brake system as set forth in claim 8 wherein said means actuated by said sensing means includes valve

means, said valve means including a housing and a rod formed with a pair of axially spaced bores and mounted for rotation within said housing between a first position and a second position, said housing formed with a first compartment in fluid communication with said second pressure chamber, a second compartment in fluid communication with said brake control unit and a third compartment in fluid communication with said first pressure source, said rod being formed and mounted in said housing to provide fluid communication between said first compartment and said third compartment only in the first rod position and to provide fluid communication between said first compartment and said second compartment only in the second rod position.

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PLAINTIFF'S EXHIBIT 11

May 14, 1968 F. A. PERRINO 3,383,482

SWITCH DEVICES

Original Filed Nov. 2, 1964 7 Sheets-Sheet

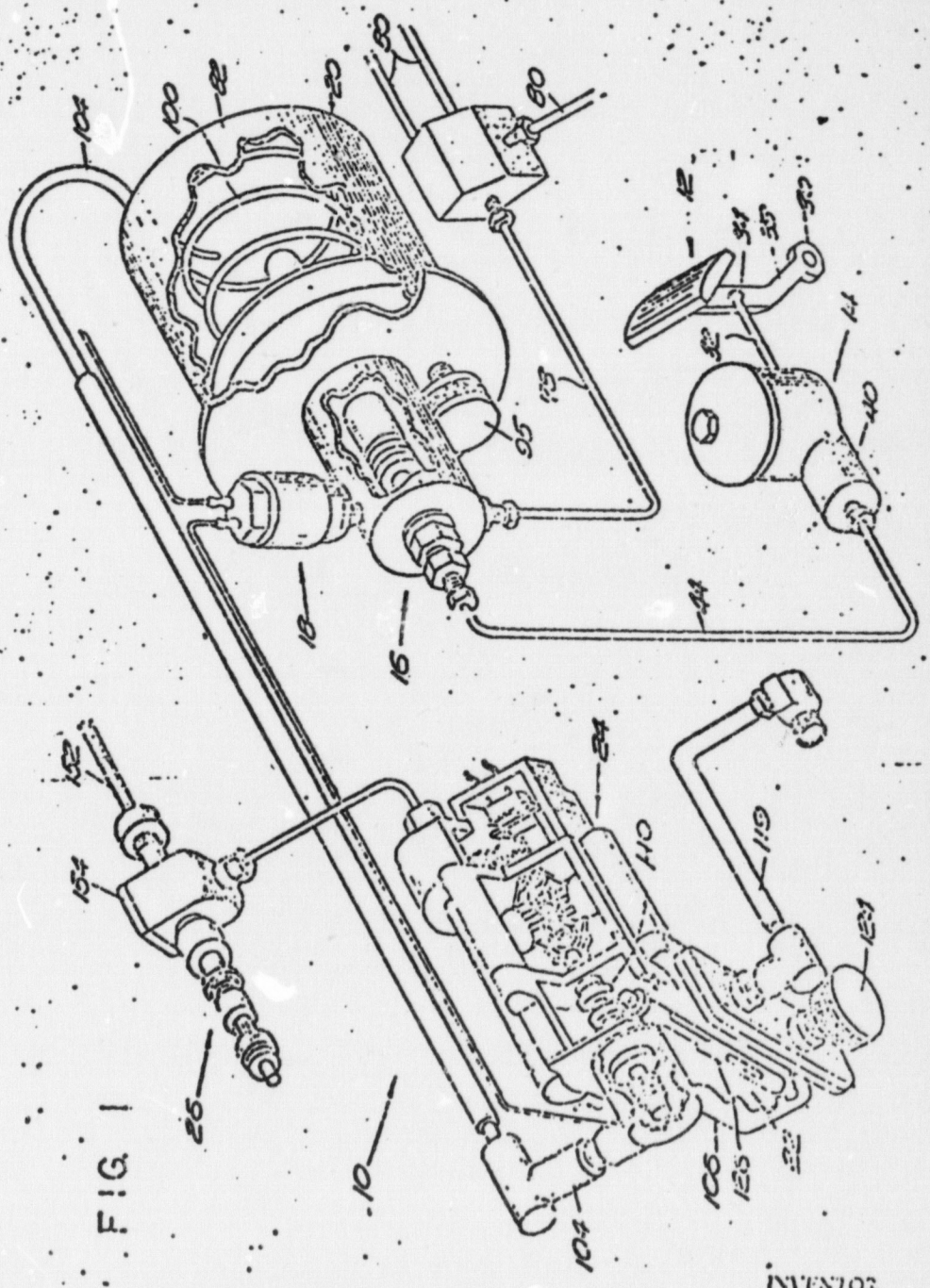


FIG. 1

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SWITCH DEVICES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 2

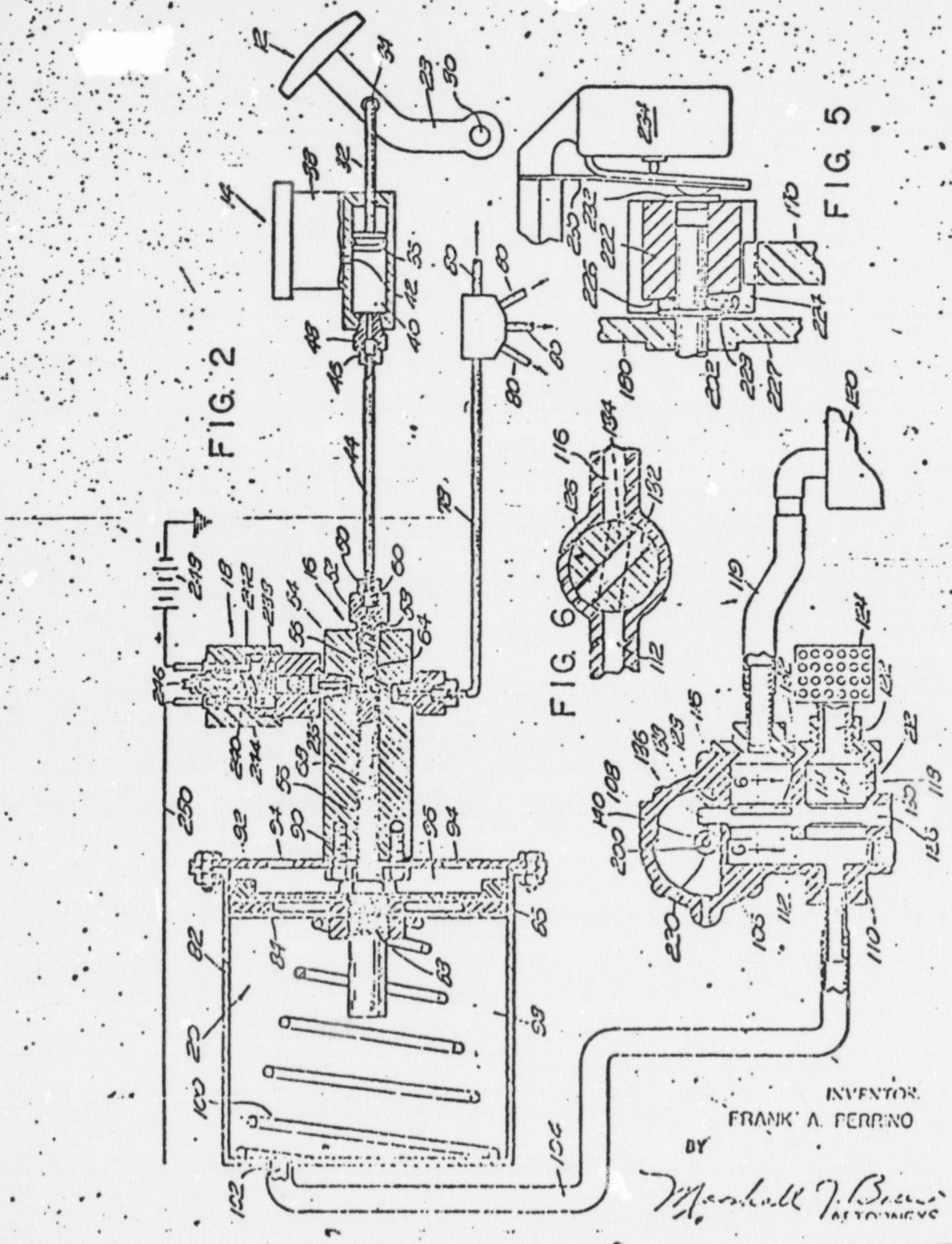


FIG. 2

FIG. 5

FIG. 6

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SWITCH DEVICES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 3

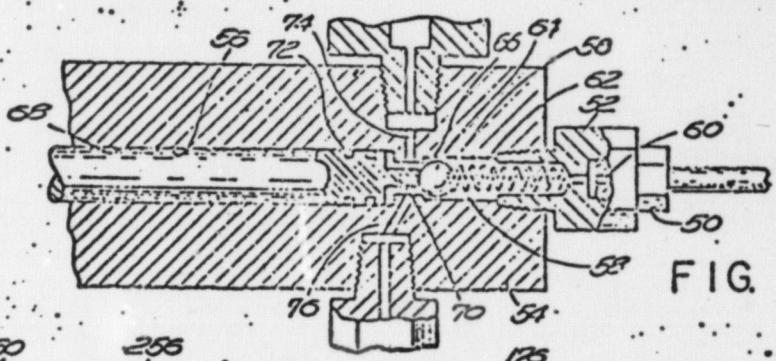


FIG. 3

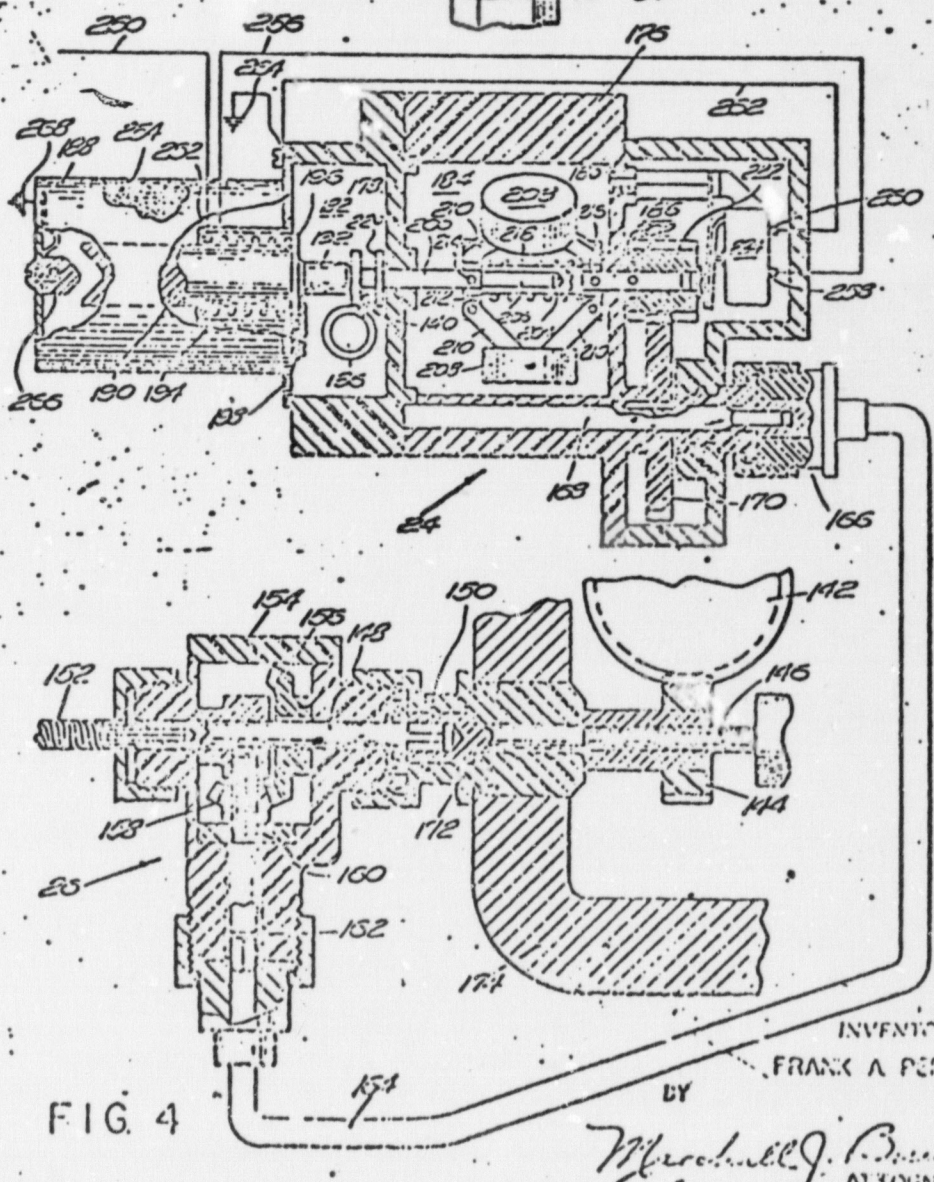


FIG. 4

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Original Filed Nov. 2, 1964 7 Sheets-Sheet 4

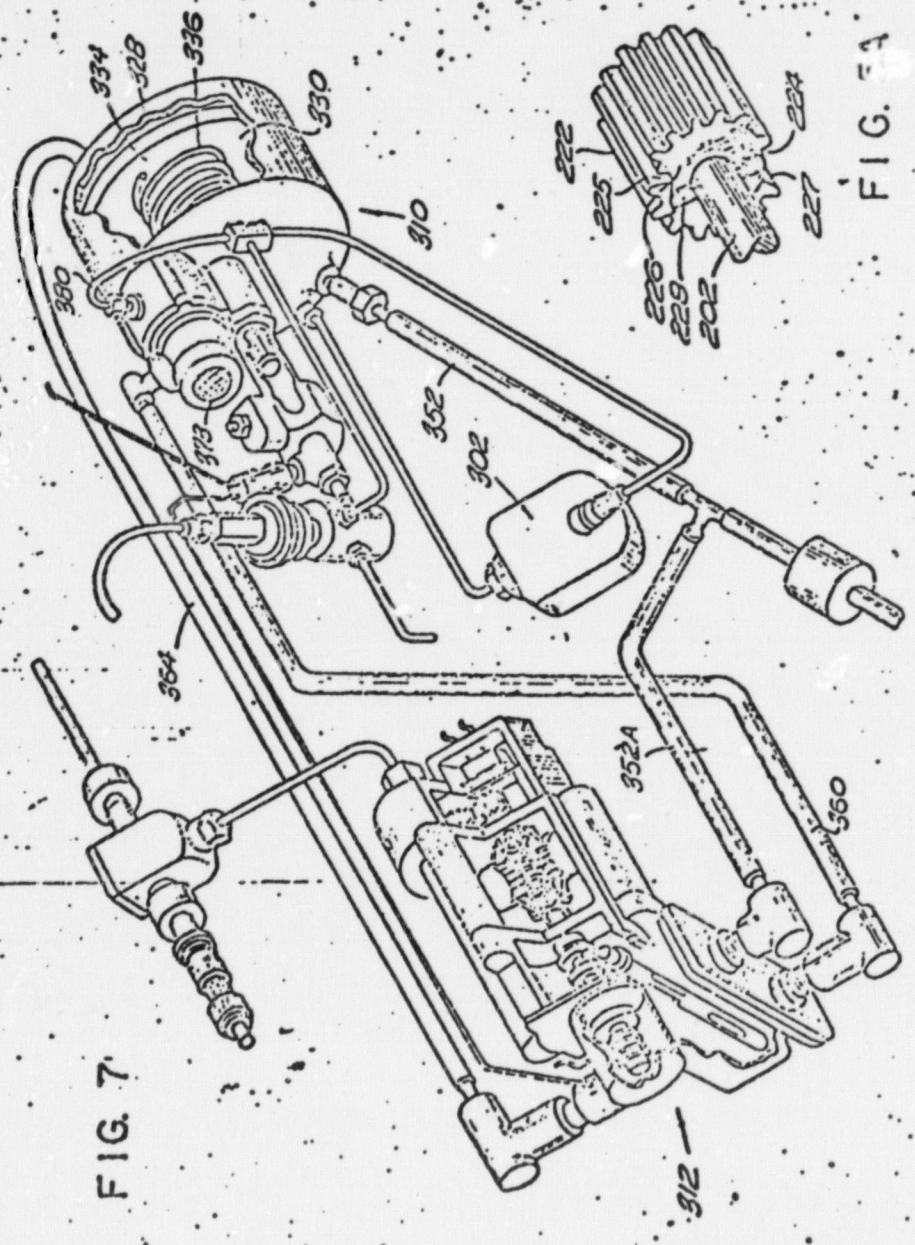


FIG. 7

FIG. 7A

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Original Filed Nov. 2, 1964 7 Sheets-Sheet 5

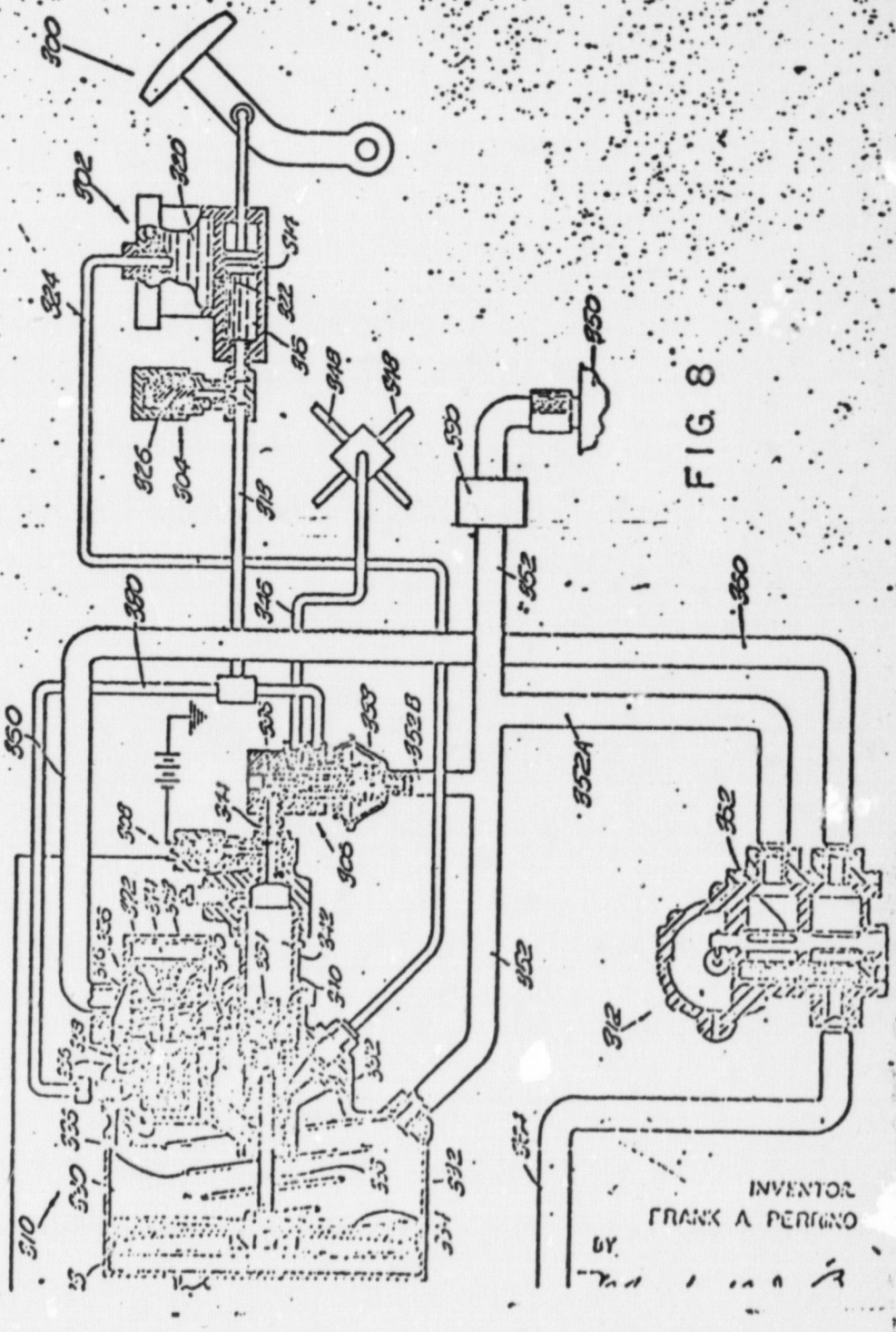


FIG. 8

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SWITCH DEVICES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 6

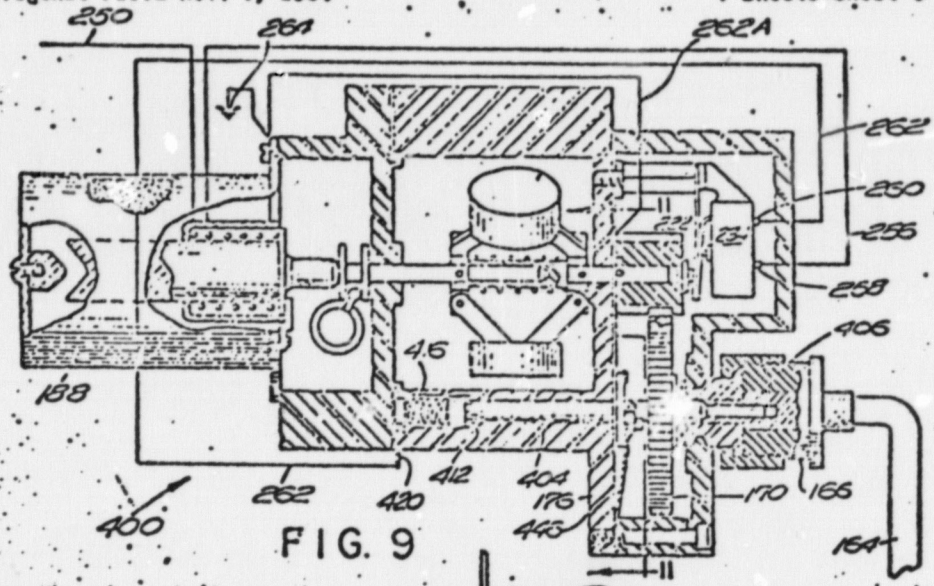


FIG. 9

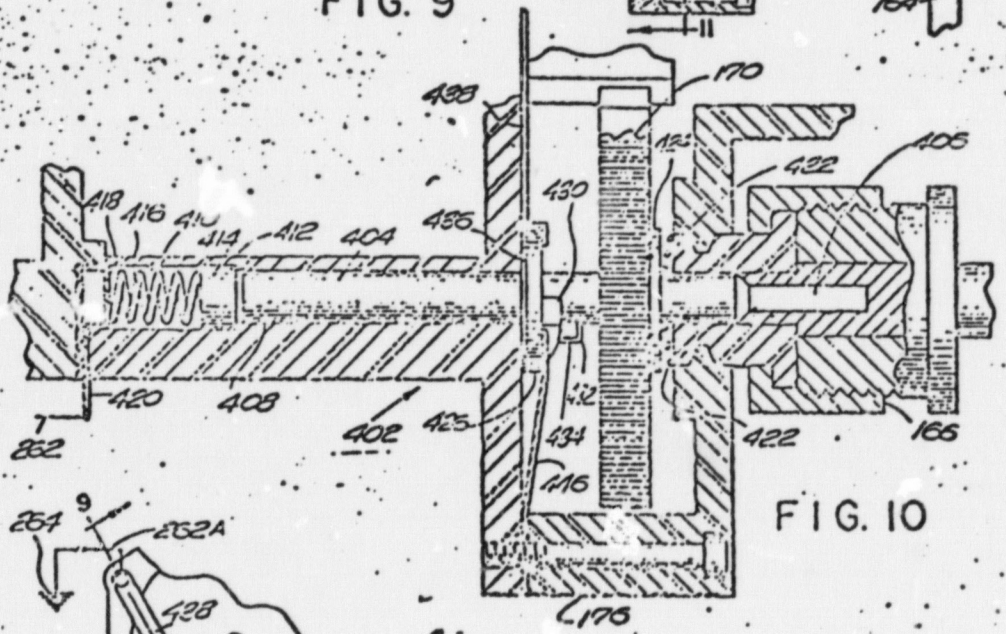


FIG. 10

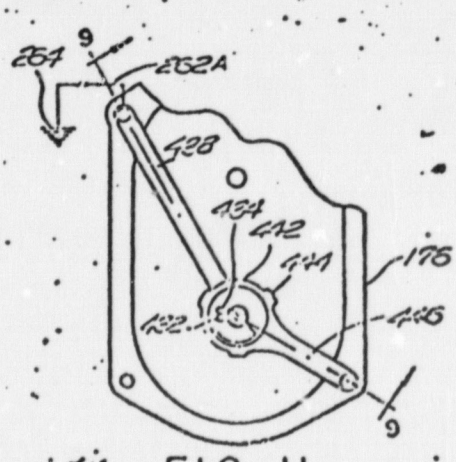


FIG. 11

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[PX 11, p 7]

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SWITCH DEVICES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 7

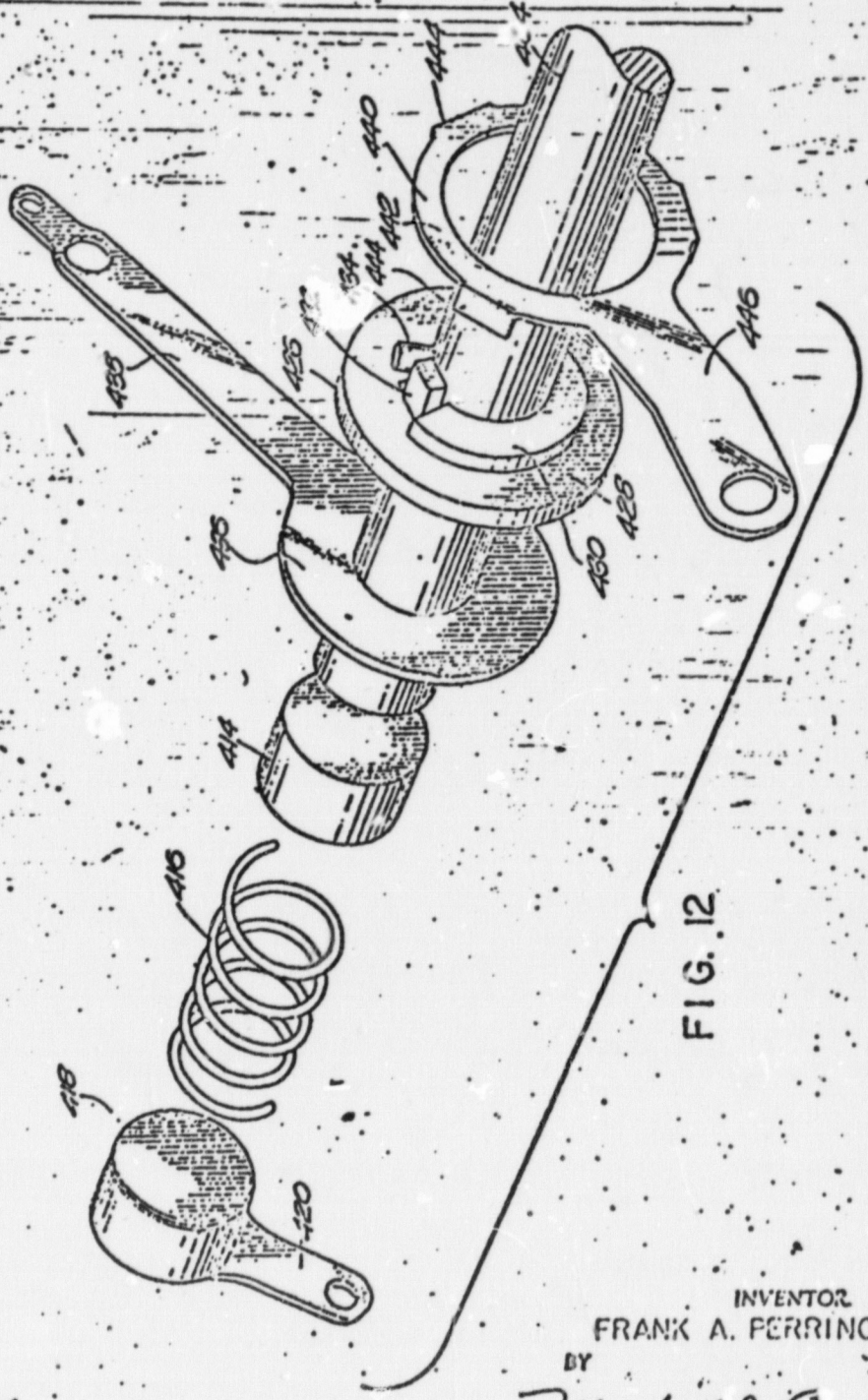


FIG. 12

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[PX 11, p 8]

United States Patent Office 3,383,482 Patented May 14, 1968

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3,383,482
SWITCH DEVICES

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Original application Nov. 2, 1964, Ser. No. 409,350, now Patent No. 3,325,236, dated June 13, 1967. Divided and this application Aug. 5, 1966, Ser. No. 591,290

7 Claims. (Cl. 200—61.39)

This is a division of application Ser. No. 409,350, filed on Nov. 2, 1964, and entitled, Braking System and Anti-Skid Means Therefor.

The present invention relates generally to braking systems for automotive vehicles and the like and is particularly concerned with the provision of novel and improved anti-skid means.

It is well known that a skid condition may be encountered by a moving vehicle where the brakes of the vehicle are applied so hard that the vehicle wheels tend to lock. As to obviously, other factors affect the commencement and existence of a skid condition, such as the condition of the surface on which the vehicle is moving, the speed at which the vehicle is traveling, etc. However, it is possible for a vehicle to commence a skid even where the surface on which the vehicle is traveling is not wet or icy and even where the vehicle is traveling at moderate speeds, if the vehicle brakes are applied too hard. Since all control of a vehicle is lost while the vehicle is in a skid, thus increasing the danger of serious accident and personal injury, it is obviously highly desirable to prevent or reduce skidding while at the same time enabling the vehicle to come to a complete stop as quickly as possible.

It is therefore a primary object of the present invention to provide an improved braking system wherein operation of the brakes is under control of the vehicle operator until a skid condition commences to exist, at which time anti-skid means automatically take over complete control of the brakes and maintain said control until danger of skidding no long exists, whereupon the brakes are returned to normal control by the operator.

Another object of my invention is the provision of efficient and sensitive control means for the various operations of the system, which have usefulness in other types of systems as well.

Another object of my invention is the provision of novel and improved anti-skid means equally adaptable to so-called power brakes, as well as nonpower brakes.

A further object of my invention is the provision of anti-skid means of the character described wherein the vacuum system of the vehicle is utilized, but wherein the use of vacuum in no way interferes with control of the vehicle brakes by the operator.

A further object of my invention is the provision of anti-skid means that is readily and easily adaptable to existing vehicles.

Another object is the provision of a novel and improved reverse cut-off switch that automatically renders the anti-skid means inoperative when the vehicle moves in reverse.

A further object of my invention is the provision of a mechanical system for automotive vehicles and the like which are durable and effective in operation and which are feasible and practical both from an assembly and manufacturing standpoint.

Other objects, features and advantages of my invention will be apparent as the description thereof proceeds when considered in connection with the accompanying drawings, wherein:

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FIG. 1 is a diagrammatic view, in perspective, of my anti-skid means in combination with a nonpower brake assembly utilized in my nonpower unit, with portions being broken away for purposes of illustration;

FIG. 2 is an elevational view, in section, of a portion of the overall assembly illustrated in FIG. 1;

FIG. 3 is an enlarged fragmentary sectional elevation of a portion of the brake-actuating means illustrated in FIG. 1;

FIG. 4 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid unit and the drive means therefor;

FIG. 5 is an enlarged sectional elevation of the cam gear and solenoid-actuating switch which form a part of the anti-skid unit;

FIG. 5A is an enlarged fragmentary perspective view of the cam gear;

FIG. 6 is an enlarged section taken on line 6—6 of FIG. 2;

FIG. 7 is a diagrammatic view, in perspective, of my anti-skid means in combination with a power brake unit, portions being broken away for purposes of illustration;

FIG. 8 is an elevational view, in section, of the power brake assembly illustrated in FIG. 7;

FIG. 9 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid means and the reverse cut-off switch in combination therewith;

FIG. 10 is an enlarged fragmentary sectional elevation of the reverse cut-off switch per se;

FIG. 11 is a fragmentary section taken on line 11—11 of FIG. 9; and

FIG. 12 is an enlarged exploded perspective view of the reverse cut-off switch per se.

According to one aspect of my invention, a sensing means for sensing when a skid condition begins is caused to activate a wheel speed-sensitive device that takes over control of the brake pressure until the skid condition ends. Advantageously, the sensing means is an inertia-responsive means that senses a sudden change in the speed of rotation of a wheel at the commencement of skid conditions. This inertia-responsive device does not control the brake pressure itself by repeated action during the skid condition, as has been attempted by the prior art. Instead the control of the brake pressure is taken over completely by a governor driven dependently with the wheel, advantageously a mechanical governor. The governor then controls the brake pressure in a hunting or modulating manner, for an extended period, until such control of the brakes is ended by a reset means that responds to slowing of the vehicle to a predetermined speed. Preferably, the reset means is a brake-pressure-sensitive device which senses reduction of brake pressure to a predetermined level which the governor can produce only when it slows to the predetermined speed.

This separation of the initiating and controlling functions is an extremely important feature. The initiating device must necessarily have a restraint to prevent operation when the vehicle merely decelerates normally. I have realized that such a restraint prevents the device from being able to effectively regulate the brake pressure due to lack of sensitivity. The governor, after it takes over control, is free of such restraint and therefore can have much greater sensitivity with the result that the brake pressure can be much more accurately regulated.

The reset device is also of critical importance because at low speeds, e.g. below 10 m.p.h., the governor forces can become so diminished that the governor can behave entirely as if the vehicle were skidding, though not, thereby totally relieving the brake pressure. The reset device can act before this occurs, whereby normal control is

[PX 11, p 8]

minimum brake pressure at which skidding of the vehicle will occur on ice, generally between 100–150 p.s.i. for conventional automobiles, can be utilized to determine the setting for the brake pressure-sensitive reset device. Thereby, only at the time when skidding is no longer possible will control be taken away from the anti-skid unit to return the brake system to normal operation. This pressure-sensitive device, particularly when taking the form of a brake fluid pressure-sensitive switch, is useful with anti-skid systems generally, though of particular usefulness in the system described above.

Advantageously, the brake pressure-sensitive switch operates an electrical system, preferably a solenoid, to restore normal brake operation, and it can be employed to bring the anti-skid unit into readiness whenever the operator applies his brakes, as well as to stop the anti-skid unit from acting when skid conditions have ended.

Another aspect of the invention concerns the control of the governor. In normal operation of the brakes, the governor is prevented from influencing the brakes, but once activated, e.g. by the inertia-sensitive device, it is made continually to act, even when the vehicle wheel speeds up after the brakes have been initially relieved. A controlling or restraining member, preferably a solenoid, is employed to achieve this operation, with the controlling member held steadily from interfering with the governor until the reset device operates. In the case where a rotary weight mechanical governor is employed, a plunger can normally restrain the governor and a solenoid, initially under the control of the inertia-responsive device, can pull back the plunger to free the governor, whereupon the plunger can operate a switch to maintain the solenoid energized, and at the end of skid conditions the reset device can de-energize the solenoid to once again restrain the governor.

Another aspect of the invention is a fluid valving system that responds to the governor to increase and decrease brake pressure. Advantageously, the valve, discharging to a brake control conduit, is moved by the governor alternately between two positions to connect two passages for fluid at different pressures to the conduit. This causes the brake pressure to rapidly vary in a modulating or hunting manner, to keep the brake pressure at maximum values while preventing skidding. The mean about which the pressure modulates changes as the slipperiness of the road changes, due to the effect such road changes have upon the governor.

For use in a closed-circuit hydraulic system, the direct control of the brakes is achieved, according to the invention, by an isolation valve that continually seals the wheel brake from the control of the operator, and a variable volume chamber, e.g. piston and cylinder, for relieving and reapplying brake pressure to the wheel, the piston modulating about a mean determined by the modulating pressure conditions of the brake control conduit. Advantageously, the piston, at the position where it allows full pressure, can operate the isolation valve. Then the piston being spring loaded to that position can also perform a pressure-limiting function, limiting brake pressure to e.g. 1,000 p.s.i., though the operator is capable of applying twice that much. In normal operation the brakes function as well at 1,000 p.s.i. as above, and by so limiting the brake pressure, less pressure is present to be relieved when a skid occurs, which enables faster action and allows the relief chamber to be smaller.

In other brake systems the brake control conduit can control a power unit, or in the case of air brakes, it can

ly to FIGS. 1–6 thereof, a closed hydraulic circuit braking system employing my novel and improved anti-skid means in combination with a so-called nonpower brake will be described. The over-all system, generally designated as 10, comprises brake pedal 12, master cylinder 14, brake-relieving device 16, hydraulic pressure switch 18, relief actuator means 20, conduit valve means 22, skid-sensing and control means 24, and anti-skid unit drive means 26.

Referring to FIGS. 1 and 2, the brake pedal 12 is of conventional construction and comprises bell crank 28 pivotally connected to a suitable support on the vehicle (not shown) at 30 whereupon depression of pedal 12 causes bell crank 28 to rotate in a counterclockwise direction around pivot point 30, thus actuating connecting rod 32 pivotally connected to bell crank 28, as at 34. Rod 32 carries at its other extremity a piston 36 which functions to pump oil or other suitable fluid from master cylinder 14, all in a well-known manner. Specifically, master cylinder 14 has an upper reservoir 38 which communicates with lower chamber 40 by means of opening 42 whereupon a full supply of oil is always present in the chamber 40 to be pumped by actuation of pedal 12 and piston 36.

Fluid pumped from master cylinder 14 passes through a conduit 44 connected at one of its extremities by suitable couplings 46, 48 to chamber 40 of master cylinder 14 and at its other extremity by a suitable coupling 50 to the brake-relieving device generally designated at 16 which incorporates an isolation valve. More specifically coupling 50 is connected to a second coupling 52 which is threadedly attached to block 54, said block having bore 56 extending longitudinally therethrough into communication with aligned longitudinally extending bores 5 and 60 in couplings 52 and 50, respectively.

As will be seen most clearly in FIGS. 2 and 3, couplin 52 is provided with an internal shoulder for receivir positioning spring 62 which positions isolation valve ba 64 adjacent valve seat 66, tending to close with slig force the bore or passageway 56. During normal oper tion of the vehicle, ball 64 is maintained in its unseate position, as illustrated in FIGS. 2 and 3, against the actio of spring 62, by means of spring-loaded actuator pist 68 having a reduced extension 70 in engagement wi ball 64. Suitable O-rings or the like 72 are provided insure a tight sliding fit between piston 68 and bore whereby said piston effectively blocks further flow fluid through bore 56.

Block 54 is provided with a pair of oppositely posed passageways 74 and 76, it being noted that passageways communicate with bore 56 just behind va seat 66, whereupon when ball 64 is in its closed posi in engagement with valve seat 66, fluid in conduit from the master cylinder will be blocked before re ing the passageways 74 and 76. Passageway 74 conmu cates with hydraulic pressure reset switch 18 for rea hereinafter to be described, while passageway 76 communicates by means of suitable couplings with co 78, which in turn communicates with the hydraulic b lines 80 leading to the wheel hydraulic actuator cylin (not shown) of the vehicle to actuate the brakes in known fashion. Thus it will be seen that with the in the positions illustrated in FIGS. 2 and 3, and ticularly with piston 68 in the position illustrated, pedal 12 controls actuation of the vehicle brakes in a direct line. This is the normal operating condition the vehicle.

The actuator for the relieving device 16, generally designated at 20 (FIGS. 1, 2) controls the move-

[PX 11, p 10]

threadedly connected, as at 88, to the aforementioned small hydraulic piston 68, whereupon movement of large piston 84 causes corresponding movement of piston 68. Housing 82 is secured to block 54 by any suitable means, such as screws 90, and the adjacent end wall 92 of the housing 82 is in free communication with atmosphere, as by ports 94. Suitable air filters, such as shown at 95 in FIG. 1, may be associated with the ports 94. It will thus be seen that the chamber 96 located on one side of large piston 84 is always at atmospheric pressure, while chamber 98, located on the opposite side of piston 84, may either be at atmospheric pressure or it may be vacuum charged, as will hereinafter be described in more detail. When the chambers 96 and 98 are both at atmospheric pressure, spring 100 resiliently urges piston 84 to the position illustrated in FIG. 2, it being seen that the piston 84 has moved to the right as far as it can go. In this position, the piston 68 has also been axially moved to the position illustrated, in which position extension 70 (FIG. 3) has engaged ball 64 to unseat it from valve seat 66. Thus, where chambers 96 and 98 are both at atmospheric pressure, the vehicle brakes are under the complete control of the brake pedal 12. However, when chamber 98 is vacuum charged, the pressure differential between chambers 96 and 98 is sufficient to overcome spring 100 and thereupon piston 84 will move to the left, viewing FIG. 2, thus causing corresponding movement to the left of piston 68. This in turn enables ball 64 to seat against valve seat 66, by means of spring 62, thus blocking the bore 56 and preventing actuation of the brakes by means of brake pedal 12. As pistons 84 and 68 move further to the left due to the presence of a vacuum or even a partial vacuum in chamber 98, hydraulic pressure in lines 78 and 80 and in the spring-loaded brake actuators associated with the vehicle wheels will be relieved, due to the fact that the fluid is now free to back up into that portion of bore 56 that has been vacated by movement of piston 68 to the left. It will therefore be seen that by introducing a vacuum or partial vacuum to chamber 98, fluid pressure in lines 78 and 80 will automatically be relieved, thereby relieving the brakes. By the same token, as soon as atmosphere is again introduced to chamber 98 to equalize the pressure in chamber 96, the brakes can once again be fully actuated by the pedal 12. It will be apparent that the amount of vacuum introduced to chamber 98 will determine the amount of reduction in brake pressure. More specifically, the greater the vacuum that is introduced to chamber 98, the greater will be the movement of piston 84 to the left (viewing FIG. 2) and at the same time there will be greater movement of piston 68 to the left, thus venting more of bore 56 and providing greater relief of the hydraulic brake pressure. Similarly, given a certain amount of relief due to a certain amount of vacuum, then increase in the amount of vacuum will increase the hydraulic brake pressure due to movement of the pistons to the right. The ability of my system to provide varying degrees of relief and restoration of brake pressure while the brake pedal remains isolated from the system is an important and advantageous feature.

It will be noted that during normal operation of the system, as heretofore described, i.e., where atmospheric pressure exists in both chambers 96 and 98, the piston 84 will function as a pressure limitor in the system. More specifically, since the piston 68 is resiliently held in the position illustrated in FIG. 2 by means of spring 100, it will back off when excess pressure is applied by foot pedal 12. In practice it has been found that in a conventional automobile pressure above 1,000 p.s.i. or at most 1,500 p.s.i., depending on the type of vehicle, is unnecessary for normal brake operation and hence springs 100 are preferably calibrated so as to enable ball 64 to seat when pressure in excess of that needed is introduced to the brake system. However, the springs 100 may be calibrated so as to provide a limit at any selected pressure.

The means for introducing either atmospheric pressure or vacuum to chamber 98 and for modulating the degree of vacuum therein will now be described. Referring to FIG. 2, it will be seen that cylinder housing 82 is provided with an opening 102 in communication with chamber 98, which opening communicates with a brake control conduit 104 that in turn connects with the conduit valve, generally designated at 22. More specifically, there is provided a housing 106 having an upper portion 108 which houses the skid-sensing and control means 24, and a lower portion 110 which houses the valve 22. The lower housing 110 comprises an inner chamber 112 having a partition 114 defining a vacuum passage 116 and an atmosphere passage 118. More specifically, vacuum passage 116 is in communication, by means of passage 119, with the intake manifold 120 of the vehicle. Thus, suction is constantly being applied to the passage 116. The passage 118, on the other hand, is in communication with atmosphere through port 122, to which a suitable air filter 124 may be attached, if desired. A rotary valve 126 is mounted for rotation in housing 106, it being noted that valve 126 is an elongated rod that extends vertically through upper partition 128, the aforesaid partition 114, and is journaled in bottom wall 130, the partition and wall being integral parts of a unique molded housing of antifriction plastic. The valve 126 is provided with a pair of passageways or bores 132 and 134 extending diametrically therethrough, said bores being vertically spaced from each other, and having elongated cross-sections in the axial direction, longer than the diameter of the rod. Bore 132 is associated with passage 116 and bore 134 with passage 118. The bores are angularly disposed with respect to each other, as shown most clearly in FIG. 6. Thus, with the rotary valve 126 in the position illustrated in FIGS. 2 and 6, bore 134 is positioned so as to allow free passage of atmosphere from passages 142 and 118, bore 134 and chamber 112 to brake control conduit 104, whereby atmosphere will flow into chamber 98. At the same time, due to the angular displacement of bore 132, passage 116 will be closed by valve 126, thus preventing suction from manifold 120 and passage 119 from coming into contact with chamber 112 and brake control conduit 104. It will be understood that the only access from passages 116 and 118 to chamber 112 is through valve 126 and more specifically, the bores 132, 134 thereof. Should valve 126 be rotated by means hereinafter to be described, it will be seen that communication between passage 118 and chamber 112 will eventually be blocked, thus interrupting the flow of atmosphere to brake control conduit 104, while at the same time passage 116 will be in communication with chamber 112, whereupon a suction will be applied through brake control conduit 104 to chamber 98.

It is important to note that use of a rotary valve such as valve 126 is extremely advantageous where passage of a vacuum and atmospheric pressure, or more generally two different pressures, is being controlled due to the fact that the pressure differential offers a minimum of resistance to the turning movement of the small diameter valve. This is contrasted to a reciprocal or poppet type valve wherein movement of the valve is much more resisted by pressure differential.

As hereinabefore explained and described, introduction of a vacuum or partial vacuum to chamber 98 automatically tends to release brake pressure on the vehicle wheels. It has been found in practice that an angular displacement between bores 132 and 134 of approximately 37.5° gives best results in the operation of my system, and it will be understood that rotary valve 126 may not always be positioned so as to completely open or close

closed by spring 374. Spring 376 normally urges dia- ... specifically, conduit 352H is connected to the suction con-

to be exerted on the vehicle wheels, thus providing the proper braking pressures for different road conditions. In instances where the pressure on the brake actuator is the same at the beginning of a skid (lesser initial pressure requires less relieving), the amount of relief of the brake pressure depends on the amount of travel of piston 84 to the left in chamber 92 which in turn is dependent on the amount of vacuum introduced to the chamber. Thus, where the vehicle is braked and skids on a dry surface which requires a greater brake pressure to create a skid condition, lesser movement of piston 84 to the left, and hence less vacuum, will be required to relieve the brake pressure sufficiently to prevent the vehicle wheels from locking, this pressure being referred to as "release" pressure. Conversely, where the vehicle is braked on a slippery surface such as ice, a lesser brake pressure is required to create a skid condition. Hence the valve 125 must open suction passage 116 to the brake control conduit 104 for a longer initial period to provide greater vacuum to chamber 92 in order to move piston 84 sufficiently to the left so that sufficient relief of the brake pressure will be effected to prevent the wheels from locking.

In accordance with the invention after release pressure is attained, with corresponding movement of the piston then the governor causes the degree of vacuum in conduit 104 to modulate, which modulates the brake pressure through a narrow range of intermediate pressures above and below the release pressure, without ever restoring the normal, higher brake pressure until the skid condition ends.

To thus operate the valve, the upper extremity of valve 126 (FIG. 2) is provided with a reduced portion 136 that extends upwardly through partition 120, said portion 136 having a pin 138 extending therethrough to which is attached a circular projection 140, it being seen that as a longitudinal thrust is exerted against projection 140, the valve 126 will be caused to rotate.

The means for controlling movement of valve 126, namely, the skid-sensing and governor control means and the drive means therefor, will now be described. Referring to FIG. 4, skid-sensing and governor control means is designated generally at 24, and the drive means therefor is designated generally at 26. Referring first to the latter, the vehicle drive shaft is shown at 142, it being understood that the drive shaft is directly connected to the wheels of the vehicle (not shown) whereupon when the vehicle wheels lock, the drive shaft also locks. The drive shaft drivingly engages speedometer gear 144, which in turn is fixed to shaft 146, which in turn is drivingly connected to shaft 148 by means of a coupling shown at 150, said shaft 148 connecting with speedometer cable 152. Gear box 154 through which shaft 148 passes and in which the shaft 148 is suitably journaled, houses drivingly engaged bevel bears 156, 158, it being understood that bevel gear 156 is keyed to shaft 148 and bevel gear 158 is keyed to shaft 160, whereupon rotation of shaft 148, transmitted thereto by the vehicle drive shaft 142, is transmitted to shaft 160. Shaft 160, in turn, is drivingly interconnected, as at 162, to flexible shaft 164, said flexible shaft being connected, as at 166, to the unit 24, and said flexible shaft being drivingly interconnected with a shaft 168 which has fixed thereto a drive gear 170. It will therefore be seen that the gear 170 is drivingly interconnected with the vehicle drive shaft 142, which in turn is connected to the vehicle wheels. Thus, rotation of the vehicle wheels will cause corresponding rotation of gear 170, and, conversely, sudden slowing of the vehicle wheels, such as would occur where the vehicle approaches a skid condition by the operator applying more brake pressure than is required for a particular road surface, will cause not only the main drive shaft 142 to suddenly slow, but also drive gear 170. Gear box 154 and adjacent

easily and readily adapted to a conventional vehicle for driving the sensing and governor control means now to be described.

The unit 24 (FIGS. 1, 4) comprises a molded plastic housing 176 having a pair of spaced partitions 178, 180 therein defining chambers 182, 184 and 186. A solenoid 188 is secured to housing 176 adjacent chamber 182 by any suitable means, the solenoid 188 having a plunger 190 with a reduced extension 192 resiliently urged into chamber 182 by means of spring 194. When solenoid 188 is not energized, spring 194 bears against flange 196 affixed to plunger 190 to resiliently urge extension 192 into chamber 182. Suitable stop means 198 mounted on the interior of chamber 182 engages flange 196 to limit resilient axial movement of plunger 190 and extension 192 at a predetermined point, for reasons hereinafter to be explained more fully. Journaled in the partitions 178 and 180 are a pair of aligned shafts 200 and 202, respectively. Shaft 202 is provided with an enlarged portion 204 having an internal bore for slidably receiving in telescoping relation a reduced extension 206 of shaft 200. Attached to shafts 200 and 202 is a governor assembly comprising a plurality of weights 208, each pivotally connected to a pair of links 210, which links in turn are pivotally connected to arms 212. The arms 212 located on one side of the weights 208 are affixed to shaft 200 as at 214, while the arms on the other side of the weights are affixed to shaft 202 as at 216. As hereinbefore pointed out, shaft 202 is journaled for rotation in partition 180 but is incapable of longitudinal movement. Shaft 200, on the other hand, is drivingly engaged with shaft 202 so as to rotate therewith, but said shaft 200 is axially slidable with respect to the said shaft 202. Since the governor assembly is affixed on one side to shaft 202 and on the other side to shaft 200, it will be seen that the speed of rotation of the shafts will determine their longitudinal position relative to each other. Spring 218 resiliently urges the governor assembly to its collapsed position wherein shafts 200 and 202 are axially extended with respect to each other, but as the rotational speed of shafts 200 and 202 increases, centrifugal force urges the weights 208 outwardly, thus moving shaft 200 toward the right, viewing FIG. 4, against the action of spring 218.

Fixed to shaft 200 at the extremity thereof located in chamber 182 is a spool member 220, which spool member is in engagement with projection 140 carried by the reduced upper extension 136 of rotary valve 126. Thus, axial movement of shaft 200 causes like movement of the spool 220 which in turn carries therewith projection 140 captured between the opposed walls of the spool to thereby impart rotary movement to valve 126.

The means for imparting rotational movement to shafts 200 and 202 and the governor assembly associated therewith comprises a cam gear 222 (FIGS. 4, 5 and 5A), which cam gear is part of the drive train, in driven engagement with gear 170. The cam gear 222 is mounted on shaft 202 and is located in chamber 186. The cam gear is slidably mounted on shaft 202 and is in driving engagement therewith by means now to be described. A recess 224 having inclined or helical surface 226 is provided at the end of the cam gear adjacent partition 180, and a projection or pin 228, carried by shaft 202, extends into said recess in engagement with the cam surface 226. At either end of the recess is a stop surface, at the deeper end stop surface 227, adapted to engage pin 228 for accelerating the governor and at the shallower end stop surface 225 for engaging pin 228 to decelerate the governor. A leaf spring 230 mounted to the housing 176 by any suitable means engages a button 232 which is press fitted into the central bore of cam gear 222 to normally urge the latter axially toward partition 180, hold-

430 mounted thereon, and further having an outwardly 75 whenever the vehicle is moving in reverse.

[PX 11, p 12]

wheels upon the presence of a skid condition, the rotational inertia of shaft 202, especially due to the governor weights 205, causes shaft 202 to turn ahead of gear 222, causing pin 223 to leave stop surface 227 and move to top surface 225. This slides gear 222 on shaft 202 in a direction away from partition 180 to activate governor control.

It is important to note that solenoid spring 194 is stronger than governor spring 218, whereupon when solenoid 188 is not energized, plunger extension 192 bears against spool 220 to restrain same in the position illustrated in FIG. 4 wherein the rotary valve 126 is in the position illustrated in FIG. 2, namely, the position where atmosphere is free to pass through the valve to conduit 104 and then to chamber 98 and wherein at the same time suction from the intake manifold is blocked from communication with conduit 104. This is the normal operating condition of the system wherein the balanced pressure on opposite sides of actuator piston 84 enables spring 100 to position piston 84 and piston 68 as illustrated in FIG. 2. In this position, ball valve 64 is unseated to allow fluid pressure to pass directly to the brake line whereupon operation of the brakes is under direct control of brake pedal 12. This condition will remain so long as solenoid 188 (FIG. 4) is not energized. Energization of solenoid 188 is accomplished in the following manner. As previously explained, sudden slowing of gear 170 due to sudden slowing of the vehicle wheels as a result of a skid condition causes cam gear 222 (FIGS. 4, 5, 5A) to move axially on shaft 202 away from partition 180 against the action of spring 230. This movement is sufficient to cause actuation of a switch 234 suitably mounted in chamber 186. At the same time, the initial movement of the brake pedal and the resultant increase in fluid pressure in the system, causes hydraulic switch 18 (FIG. 2) to become closed. Hydraulic switch 18 is fully disclosed and described in copending U.S. patent application Ser. No. 203,075, filed June 18, 1962. Basically, switch 18 comprises piston 235 which has connected thereto at its upper end a metallic contact cup 236. Cup 238 is adapted to bridge contacts 240 and 242 but is normally maintained spaced therefrom by spring 244. An adjusting screw 245 is provided by means of which the tension of spring 244 may be varied, thus enabling the switch 18 to be preadjusted whereby a predetermined fluid pressure, e.g. 120–125 p.s.i., will cause piston 235 to move against the action of spring 244 sufficiently to bridge the contacts 240, 242. Actuation of switch 18 in the aforedescribed manner enables current to flow from battery 248 through contact 242, contact cup 236, and contact 240 to line 250 to the solenoid coil positive terminal 252 (FIG. 4). The voltage is applied through the solenoid coil 254 and then through line 256 to terminal 258 on switch 234. When switch 234 is open, the circuit to ground is not complete, and solenoid 188 will not be energized. However, upon actuation of switch 234 in the manner aforedescribed, the circuit to ground is completed through terminal 260, line 262 and ground connection 264. It is important to note that the circuit is such that although it is essential for hydraulic pressure switch 18 to be closed in order for solenoid 188 to be energized, closing of switch 18 alone is not sufficient, but rather switch 234 must also be actuated in order to complete the circuit. This is important because under normal operating conditions of the system sufficient pressure will normally be introduced to switch 18 by the foot pedal to actuate same, but it is essential that solenoid 188 not be energized under such normal condition.

Energization of solenoid 188 accomplished by rightward movement of the cam gear (FIG. 5) closing switch 234, causes plunger 190, 192 to be retracted against the operated plunger together can be regarded as an inertia-responsive means for initiating the action of the governor.

As soon as the solenoid plunger has been retracted, due to the sudden slowing at the commencement of a skid condition, the governor assembly takes over control and, as it is stopped by stop surface 225 of the cam gear, spring 218 will cause the governor assembly to commence to collapse, thus moving shaft 200 and spool 220 axially to the left, when viewing FIG. 4. This movement of spool 220 causes corresponding rotary movement of valve 126, thus shutting off the passage of atmosphere to brake control conduit 104 and simultaneously effecting communication between vacuum passage 119 and conduit 104. This results in suction being introduced to chamber 98 which in turn overcomes the action of spring 100 to cause pistons 84 and 68 to move to the left, viewing FIG. 2. This movement of piston 68 enables ball valve 64 to close due to the action of spring 62, thus preventing the introduction of further fluid pressure to the system via brake pedal 12. Further leftward movement of piston 68 enables the fluid in lines 78 and 80 to back up, thereby relieving the pressure until the vehicle brakes begin turning, the duration that the suction is applied, and hence the degree of vacuum and the amount of pressure relief depending on the slipperiness of the road. As soon as the brake release pressure is achieved, the vehicle wheels commence to rotate, whereby drive shaft 142 through the aforedescribed gearing once again causes gears 170 and 222 to rotate, it being apparent that this renewed rotation of shafts 200 and 202 will cause the governor assembly to once again turn, and due to the centrifugal force created, to retract shaft 200 and spool 220 to once again rotate valve 126 to the positions illustrated in FIGS. 2 and 4, where once again atmosphere is connected to chamber 98, thus enabling brake pressure to be restored.

But, it is very important for my present invention that the solenoid is not de-energized and the governor returned to the restraint of the inertia-responsive device. Rather, the governor remains free to act. It leaves the valve 126 open to atmosphere 118 for an instant until the degree of vacuum is lessened enough to cause slight rightward movement of the piston 84 to cause the brake pressure to slightly exceed release pressure and a skid again commences. Again the governor is slowed by the drive train and again it opens the valve 126 to suction 116, and this cycle repeats again and again in an extremely rapid hunting or modulating action with the gas pressure in the conduit 104 never reaching either atmospheric or engine vacuum level, but rather modulates at intermediate levels. In dependent fashion, the hydraulic brake pressure modulates narrowly about the release pressure as a mean, without returning to normal pressure. When the skid condition ends, the governor opens the valve continually to atmosphere and pistons 84 and 86 move fully to the right to once again unseat insolation valve 64 and restore normal brake operation. The faster the vehicle travels, the faster does this action occur, but the action is sufficiently fast down to speeds on the order of 10 m.p.h. and even lower, to rapidly pump and release the brakes. It is another very important aspect of the invention that the governor relinquishes control of the brake pressure before the vehicle stops.

In order to effectively carry out this modulating action, I have found it to be critical, in the preferred construction, that the solenoid 188 remain energized or that the plunger in some other way remain retracted until the vehicle slows to a predetermined slow speed, e.g. 5 or 10 m.p.h., depending on the vehicle. Since closing of switch 234 to ground has given it to ground that energizes solenoid but is only momentary, holding means are

[PX 11, p 12]

the two chambers causing said second piston to move in a direction to urge said first piston to move in the first direction, means for biasing said first piston in a second direction, and means for selectively applying and relieving fluid pressure to said brake actuators whereby said pressure for said brake actuators is alternately applied and relieved while control of the amount of fluid pressure applied to the brake actuators is removed from the operator.

to achieve a skid condition, e.g. set to 85-90 p.s.i. produced by the governor at 5 or 10 m.p.h.

The aforesaid holding means for the solenoid comprise a terminal 266 having a connection to ground as at 262, which terminal is engaged by plunger 190 when the latter is retracted in order to complete the circuit to ground independent of switch 234. Thus, although closing of switches 18 and 234 is essential in order to energize solenoid 188, once the solenoid has been energized and its plunger retracted, switch 234 may be opened without de-energizing the solenoid, and the solenoid will remain energized until switch 18 opens.

As hereinbefore stated, it will be obvious that if switch 18 were not utilized, then there would be no means present for insuring de-energization of the solenoid once the condition has been reached where a skid is no longer possible, it being obvious that if the solenoid were still energized at such low speeds, the vehicle would be without effective brakes, since the governor assembly would not exert sufficient centrifugal force to overcome the action of spring 218 and move shaft 200 and spool 220 to the rightward position (FIG. 4) wherein valve 126 enables the brakes to be controlled by pedal 12.

It is possible for the reset device to take some other form, such as a pendulum device or a vehicle inertia memory system that can detect the sl[illegible] speed of the vehicle, but brake pressure-sensitive devices, and particularly hydraulic-electric switches have important advantages in the system, particularly since they can respond whenever the skid condition has passed, regardless of the speed of the car, to enable normal operation, while offering low speed protection against the governor's cutting off the brakes completely. The adjustment of the pressure level to which the device responds can be adjusted according to the wishes of the operator and the type of road conditions to be expected. But, in fact, a single setting, below that at which skidding on ice occurs, which is determinable for a, given vehicle, is advantageously employed to give anti-skid protection under virtually all conditions.

Any tendency for the unit 24 to erroneously sense a skid condition and actuate switch 234 in response to a rapid deceleration is overcome by use of properly calibrated weights 208 and a properly tensioned spring 220. In addition, I have found that an erroneous sensing is most likely to occur where there is a rapid acceleration of the drive shaft quickly followed by a rapid deceleration. The frictional brake or drag that is resiliently exerted on spool 220 by plunger 190 helps to overcome this problem since this action tends to reduce the torque applied to shafts 200 and 202 by the governor assembly. It has been found, however, that at high speeds the frictional engagement between plunger 190 and spool 220 results in undue wear of the parts. Therefore, plunger 190 is assisted by spring 230, and the plunger is disengaged at high speeds. To this end, stop means 192 engages flange 196 so as to limit the outward movement of plunger 190 whereupon no engagement exists between plunger 190 and spool 220 at relatively high speeds. I have realized that the crucial problem of rapid acceleration followed by rapid deceleration occurs at low speeds in the low gear range with conventional automobiles, and I have, therefore, found that it is not necessary to have plunger 190 exert a frictional drag on spool 220 and shaft 200 at relatively high speeds.

Although the operation of my invention is thought to be clear from the foregoing description, a brief resumé of the operation will now be given. Under normal operating conditions, solenoid 188 is de-energized whereby the spring-loaded solenoid plunger maintains the skid-sensing and control unit inoperative. The de-energized solenoid further insures that rotary valve 126 is in the position wherein atmosphere is introduced to chamber 98 [illegible]

84 thus enabling spring 100 to take over and move the piston 84 to the position illustrated in FIG. 2. In this position, piston 66, which is connected to piston 84 and moves therewith, engages ball valve 64 so as to unseat same against the action of spring 62. With ball valve 64 so opened, fluid pumped from master cylinder 14 by brake pedal 12 is free to pass directly to hydraulic lines 78 and 80 to actuate the vehicle brakes. The pressure-sensitive switch is adapted to be closed by a low level of brake pressure, e.g., 120-125 p.s.i., placing the anti-skid unit into readiness for operation.

Upon encountering a skid condition, i.e., a condition wherein the vehicle operator applies more brake pressure than is required for a particular road surface, the vehicle wheels slow suddenly. This causes corresponding deceleration of drive shaft 142, and, through suitable gearing, drive gear 170 and cam gear 222 also slow abruptly. The inertia of the governor assembly causes the gear 222 to move longitudinally by the cam surfaces aforedescribed, said longitudinal movement of gear 222 causing switch 234 to be closed which results in energization of solenoid 188 and the retraction of solenoid plunger 190. The second switch or terminal 266 is engaged upon retraction of the solenoid plunger to maintain the solenoid energized until switch 18 opens, even though switch 234 has now opened. Switch 18 is preset, e.g., at 85-90 p.s.i. to be opened at a brake pressure below that at which skids can occur, e.g., 100-150 p.s.i. for ice, whereupon solenoid 188 will remain energized until possibility of skidding no longer exists, and the solenoid is then de-energized to restore normal foot brake action before the governor has a chance ever to totally relieve the brakes.

Regarding the operation of the governor, once the solenoid 188 has been energized, the governor control unit takes over. More specifically, when a skid condition is approached, the sudden slowing of the vehicle wheels causes the governor assembly to collapse under the action of spring 218, thus moving shaft 200 and spool 220 to rotate valve 126, this movement now being permitted due to retraction of plunger 190. Rotation of valve 126 blocks the flow of atmosphere to chamber 98 and simultaneously allows the introduction of a vacuum to said chamber. The introduction of vacuum overcomes spring 100, thus resulting in movement of pistons 84 and 68 to the left, viewing FIG. 2. This movement closes isolation valve 64, thus preventing the application of further fluid pressure by brake pedal 12. As the valve remains in this position, the degree of vacuum increases, causing further leftward movement of piston 68 (FIG. 3), so that pressure in lines 78 and 80 and the brakes are relieved. A point is reached when the brakes are relieved sufficiently to allow the wheels to speed up from their slowed condition, the degree of relief required depending on the degree of slipperiness of the road. Then shafts 200 and 202 speed up once again, whereby the governor assembly retracts shaft 200 and spool 220 to once again rotate valve 126 to the position where atmosphere is introduced to chamber 98 and the piston 68 moves to the right and increases the brake pressure, until a part is reached where the wheels slow again and the cycle repeats. This action continues in a hunting manner so long as the vehicle is in an incipient skid, thus insuring that maximum brake pressure is continually applied to the wheel brakes short of pressure that would cause locking of the wheels. Once the brake pedal is released or the brake pressure in the system has dropped below the minimum possible value to cause a skid, hydraulic pressure which is broken the circuit to solenoid 188, whereupon the latter becomes de-energized and the brakes come again become under complete control of the operator.

It is important to note that although loss of vacuum in the vehicle system will render the anti-skid means ineffective [illegible]

[PX 11, p 14]

trolled brakes of both the powered and nonpowered types.

While a spring-loaded fly ball type of governor has been illustrated, and is presently preferred, the broad concepts of the invention include the use of other governors, all rotary weight governors, such as gravity loaded fly ball, centrifugal shaft and inertial shaft governors, as well as governors that operate through electrical action.

The concepts are likewise applicable to other brake systems including air brakes in which valve 126 could directly modulate air brake pressure. Likewise it is possible to separate the governor from the valve, let the governor actuate a switch and let the valve be actuated by a solenoid responsive to the condition of the switch.

While use with engine vacuum is very efficient and inexpensive, and unlike previous systems I have proposed, does not cause the engine to run roughly due to loss of vacuum, the invention is applicable to other sources of pressure differential, incompressible fluid as well as gaseous fluid and numerous aspects of the invention are useful with nonfluid types of brake systems.

Referring now to FIGS. 7 and 8, a slightly different form of my invention is disclosed in which the relieving device is incorporated as an integral part of a power brake unit. The system generally comprises a brake pedal 300, a master cylinder 302, an accumulator 304, a safety valve 306, a hydraulic-pressure switch 308, a power brake control unit 310, and an anti-skid unit 312.

Actuation of brake pedal 300 operates piston 314 to pump fluid from cylinder 316 to conduit 318. Master cylinder 302 further comprises a fluid reservoir 320 communicating with cylinder 316 by means of passageway 322. A conduit 324 extends from reservoir 320, the function of which will hereinafter be described. Accumulator 304 is in communication with conduit 318 and comprises a spring loaded piston 326 and acts as a cushion and shock absorber for the fluid system where variations in fluid pressure occur.

During normal operation of the vehicle with which the power brake unit is associated, and with the brakes in their non-actuated position, the power brake unit 310 is in the position illustrated in FIG. 8. In this position, chambers 328 and 330 of cylinder 332 are both under vacuum, thereby equalizing the pressure on opposite sides of piston 334. This enables spring 336 to urge the piston 334 to the leftward position illustrated, said piston 334 being connected by means of rod 338 to a brake actuating piston 340. Piston 340 is slidably mounted in cylinder 342, said cylinder having an outlet conduit 344 leading to safety valve 306. Conduit 344 is also in communication with hydraulic pressure switch 308.

A supply of hydraulic fluid is located in cylinder 342 whereupon movement of piston 334 against the action of spring 336, and corresponding movement of piston 340 will force fluid from cylinder 342 under pressure through conduit 344 and with valve 306 in the position illustrated in FIG. 8 to conduit 346 and then to hydraulic lines 348 which lead to the wheel cylinders (not shown) of the vehicle's braking system. Thus it will be seen that movement of pistons 334 and 340 to the right, viewing FIG. 8, will cause operation of the vehicle brakes.

As hereinbefore indicated, under normal operating conditions of the vehicle, and with the brakes not applied, pistons 334 and 340 are in the position illustrated in FIG. 8 due to the fact that a vacuum exists in both chambers 328 and 330. More specifically, suction from the intake manifold 350 of the vehicle, passes through conduit 352 into chamber 330 and then through passageway 354, through openings 356 in diaphragm 358 to conduit 360, through rotary valve 362 of the manifold unit 312, through conduit 364 and then to chamber 328. The control unit generally shown at 366 which forms a part of power brake unit 312 is of conventional form and comprises a housing 368 having a diaphragm 358 therein. Piston 363 is connected to diaphragm 358 and is urged to the left, viewing FIG. 8, by spring 376. Piston 363 is provided with a valve 372 which is normally held open by the action of spring 376. Thus, when the parts are in the position illustrated in FIG. 8, the vacuum in chamber 330 is communicated through conduit 360 to the left side of diaphragm 358 and through valve 372 to conduit 374 and then to the right side of diaphragm 358. Thus, the vacuum on both sides of diaphragm 358 is equalized and spring 376 urges the diaphragm to the left, viewing FIG. 8, to its open position. In this position, the diaphragm engages the fixed seat 370 to block communication between passage 354 and conduit 360. The piston 363 also controls movement of poppet valve 372 normally maintained closed by spring 374. Spring 376 normally urges diaphragm 358 to its open or unseated position, as illustrated in FIG. 8. An atmosphere intake is located at 378, and introduction of atmosphere into the system is controlled by valve 372 in a manner now to be described.

As above indicated, with the parts in the positions illustrated in FIG. 8, the brakes of the vehicle are nonoperative due to the fact that suction is free to circulate through conduit 352 to chamber 330 and then through control unit 366 to conduit 360, through rotary valve 362 to conduit 364 and then to chamber 328. With vacuum on both sides of piston 334, spring 336 maintains the piston 334 and the connected brake actuating piston 340 in brake release position. Upon actuation of brake pedal 300 by the operator, pressurized fluid passes through conduit 318 to conduit 380. With the safety valve 306 in the position illustrated, flow of the fluid into the the valve is blocked, and hence the brake fluid is forced to flow through conduit 380 into the brake control unit 366. None of the fluid from the pedal is allowed to reach the brake actuators at 348.

Introduction of the pressurized fluid to the brake control unit 366 causes actuation of piston 363 to move diaphragm 358 to its closed or seated position and to simultaneously open poppet valve 372 in a conventional manner. This movement blocks communication between chamber 330 and conduit 360 and simultaneously allows air to enter through inlet 378 through the now open valve 372 to conduit 360. The atmosphere circulates around conduit 360 through valve 362 to conduit 364 and then the chamber 328. The introduction of atmospheric pressure to chamber 328 creates a pressure differential which overcomes spring 336 and moves piston 334 and piston 340 to brake actuating position. More specifically, the movement of piston 340 in cylinder 342 forces the brake fluid through conduit 344 and safety valve 306 to conduit 346 and hydraulic brake lines 348 to actuate the vehicle brakes in known fashion.

In order to insure that there is always an adequate supply of brake fluid in cylinder 342, conduit 324 connects fluid-reservoir 320 to the cylinder 342, it being important to note that the inlet 382 to the cylinder 342 is located behind the forward portion of piston 340. Piston 340 is provided with a one-way valve 384 which enables fluid to pass through the piston from left to right, viewing FIG. 8, but which blocks flow of fluid through the piston from right to left. Thus, during operation of the piston 340, the suction created thereby in cylinder 342 sucks fluid through conduit 324 from reservoir 320 into cylinder 342 on the rear side of piston 340. During the return stroke of piston 340, the valve 384 allows fluid to pass through the piston, thereby insuring that there is always a full supply of fluid in the cylinder for brake actuation.

Since the structure and operation of hydraulic pressure switch 308 and anti-skid unit 312 are identical to that already described in connection with the first form of my invention, it is not thought that further description is necessary. It will be noted, however, that where the operator applies more brake pressure than is required for a particular road surface, the vehicle wheels will lock, whereby the same structure and operational sequence will cause rotary valve 362 to be moved to a position wherein conduit 360 becomes blocked and conduit 353A becomes open to allow suction from manifold 350 to be introduced through conduit 291 to chamber 328, whereupon spring 336 will once again take over to release the brakes. The same control [illegible]

[PX 11, p 15]

In accordance with my invention it is crucial that the entire brake pressure be controlled by the anti-skid unit and that no fluid be conducted directly from the foot pedal to the wheels. Means are provided to insure that brake pressure will not be lost should there be a loss of suction. This is accomplished by safety valve 306. More specifically, conduit 352B is connected to the suction conduit 352 whereupon as long as proper suction exists in the system, piston 306 is maintained in the position illustrated in FIG. 8 against the action of spring 308. In this position, conduit 344 is in communication with conduit 346, but conduit 350 is blocked at its extremity adjacent the safety valve. Should suction be lost in the system, spring 308 will become effective to move piston 306 upwardly, thereby blocking the communication between conduits 344 and 346 and at the same time providing communication between conduit 350 and conduit 346. With the safety valve in this position, the brakes are directly controlled by operation of pedal 390 since the entire power unit is now bypassed.

A one-way check valve 309 is located in conduit 352 adjacent the intake manifold 350 since without same backfire of the vehicle engine would cause an explosive pressure to be introduced to the power unit, which would be undesirable for obvious reasons.

Referring once again to FIG. 4, it will be noted that should the vehicle move in reverse, the direction of rotation of drive shaft 142 will likewise be in reverse, thus causing reverse rotation of flexible shaft 164 and shaft 166. Reverse rotation of shaft 166 will, in turn, cause reverse rotation of drive gear 170 and cam gear 222, causing pin 223 to leave stop surface 227 and move toward stop surface 225, which, as previously described, causes gear 222 to slide on shaft 202 in a direction away from partition 180 to close switch 234 and thus activate the governor control and anti-skid means. Even though it is possible that very slow reverse movement of the vehicle might not be sufficient to cause movement of cam gear 222 to activate the governor control, the fact remains that there is always the possibility and even likelihood that reverse movement of the vehicle will activate the governor control even though no skid condition exists. Since there is no practical need for any anti-skid means when the vehicle is moving in reverse, and since the activation of these means when not needed might even result in the loss of some stopping distance, it has been found desirable to provide means for rendering the governor control and anti-skid ineffective and inoperative when the vehicle moves in reverse.

Referring to FIGS. 9 through 12, a somewhat modified anti-skid unit is shown generally at 400. Unit 400 is basically the same as skid-sensing and control means 24 hereinbefore described except that a reverse cut-off switch has been incorporated in the unit to prevent energization of solenoid 136 when the vehicle drive shaft is in reverse and flexible shaft 164 correspondingly rotates in a reverse direction. The reverse cut-off switch, shown generally at 402 (FIG. 10), comprises a shaft 404 slidably coupled to shaft 164 as at 406, in the same manner that shaft 166 was drivingly engaged to flexible shaft 164. Shaft 404, which is constructed of any electrically conductive metal, is journalled in bore 408, said bore having an enlarged end portion 410, it being understood that the larger diameter of portion 410 results in the presence of an annular shoulder 412 at the extremity of bore 408. Slidably mounted in portion 410 is a contact button 414, said button being resiliently urged toward bore 408 by means of spring 416. The opposite end of spring 416 resiliently engages a contact cup 418 having a terminal

FIGS. 9 and 10, thus causing the end of shaft 404 to extend into enlarged portion 410 and, therefore, into engagement with contact button 414, as illustrated in FIG. 9. Loosely mounted on shaft 404 is a reverse switch cam 426 comprising a circular plate 428 having a spiral cam portion 430 mounted thereon, and further having an outwardly extending pin 432 located adjacent the end of cam track 430. A cam pin 434 extends radially from shaft 404 and is adapted to cooperate with the spiral cam track 430 to determine the axial position of shaft 404. Expressed differently, when shaft 404 is rotating in a counterclockwise direction when viewing FIG. 12, cam pin 434 will engage the stop pin 432 at the low point on the spiral track 430. With the parts in this position, shaft 404 will assume the position illustrated in FIG. 9 wherein the extremity of the shaft is in engagement with contact button 412, it being understood that spring 422 is somewhat stronger than spring 416, whereupon the extremity of shaft 404 will actually enter into enlarged portion 410. Upon clockwise rotation of shaft 404, pin 434 will ride up the spiral cam track 430 until it reaches the high end of said track and engages the opposite side of stop pin 432, as illustrated in FIG. 10. In this position, shaft 404 has been moved to the right (when viewing FIG. 10) against the action of spring 422 sufficiently for the end of said shaft to become completely withdrawn from enlarged portion 410. Since the annular shoulder 412 prevents the contact button 414 from moving out of enlarged portion 410, it follows that the end of shaft 404 becomes disengaged from contact button 414, as clearly illustrated in FIG. 10. Also mounted on shaft 404 is a washer 436 having a radially extending contact arm 438, the purpose of which will hereinafter be described. As will be seen most clearly in FIG. 10, the washer 436 is mounted on shaft 404 adjacent the flat rear surface of cam 426, it being understood that shaft 404 may freely rotate within washer 436. A cam thrust washer 440 comprising a peripheral ring portion 442 and bent marginal lugs 444 resiliently engages the opposite face 428 of cam 426, whereupon to resiliently urge cam 426 into engagement with washer 436 and at the same time to impart a sufficient frictional drag on cam 426 so that movement of shaft 404 will not impart corresponding rotary movement to the cam until pin 434 moves into engagement with stop pin 432. The thrust washer 440 has a radially extending spring arm 446 that is secured to the housing 176 by any suitable means, such as by having the end of the spring arm clamped between adjacent portions of the housing, as illustrated in FIGS. 9 and 10.

The operation of reverse switch 402 is as follows. Assuming that shaft 404 is being rotated in a counterclockwise direction, which for purposes of this description corresponds to forward movement of the vehicle, the switch will, as aforestated, assume the position illustrated in FIG. 9 wherein shaft 404 is making engagement with contact button 412. With the parts in this position and assuming that a skid condition commences to exist, causing cam gear 222 to close switch 234, current will flow through line 262 to terminal 420 through contact cup 418, spring 416, contact button 414, and then to shaft 404, which as previously described, is in engagement with contact button 414. Shaft 404 is in turn in electrical engagement with washer 436 and its radially extending terminal 438, whereupon current flows through engagement of pin 434, which in turn engages cam 426 which in turn bears against washer 436. Thus, the flow of current is through terminal arm 438 and thence to line 262A, which connects to ground as at 264. This, along with line 266, completes the circuit to solenoid 136 to

10, thus interrupting the connection to ground and preventing energization of solenoid 150. It will thus be seen that whenever the vehicle and its drive shaft moves in reverse, the reverse cut-off switch 402 will automatically prevent the skid sensing and control means and the governor control from coming into play. Thus, the vehicle brakes will always be under normal and direct control by the operator whenever the vehicle is moving in reverse.

While there is shown and described herein certain specific structure embodying the invention, it will be manifest to those skilled in the art that various modifications and rearrangements of the parts may be made without departing from the spirit and scope of the underlying inventive concept and that the same is not limited to the particular forms herein shown and described except insofar as indicated by the scope of the appended claims.

I claim:

1. An electric switch movable automatically to an open position in response to rotation of a shaft in one direction and movable automatically to a closed position in response to rotation of the shaft in the opposite direction, said switch comprising a rotatable shaft, a contact button adapted to resiliently engage one end of said shaft, means mounting said shaft for axial movement between a first position wherein said contact button is in engagement with said shaft and a second position wherein it is disengaged therefrom, and cam means associated with said shaft for automatically moving said shaft to said first position when the shaft is rotating in one direction and to the said second position when the shaft is rotating in the opposite direction.

2. An electric reverse cut-off switch for use with a vehicle wheel anti-skid system for a vehicle brake system having a fluid source, a vehicle wheel brake operating actuator cylinder in fluid communication with said fluid source, and manually operable means for applying fluid pressure to said vehicle wheel brake operating actuator cylinder, said vehicle wheel anti-skid system comprising means for sensing a predetermined deceleration of the vehicle wheel, means actuated by said sensing means for alternately applying and relieving the fluid pressure applied to the vehicle wheel brake operating actuator cylinder, means actuated by said sensing means for controlling the initiation of said means for alternately applying and relieving the fluid pressure, said sensing means including a drive gear operatively connected to and driven by the vehicle wheel and a cam gear driven by said drive gear, said controlling means including a solenoid, an electrical source including an electrical ground, and switch means in series with said solenoid for actuating said solenoid when said switch means are closed, said switch means including a switch actuable by said cam gear when the predetermined deceleration of the vehicle wheel is sensed, said switch being in series with said solenoid and the electrical ground, said electric reverse cut-off switch comprising a movable electrically conductive arm, means for electrically connecting said movable arm to the electrical ground, said electric reverse cut-off switch being interposed between the cam gear actuable switch and the electrical ground, and means for electrically connecting said movable arm to said cam gear actuable switch when the drive gear is rotated in a first direction and for electrically disconnecting said movable arm to said cam gear actuable switch when the drive gear is rotated in a second direction, whereby said vehicle wheel anti-skid system may be actuated when the wheel is being rotated in one predetermined direction but not in the other direction.

3. An electric reverse cut-off switch as set forth in claim 2 wherein said movable electrically conductive arm is a shaft and is operatively connected to the vehicle wheel and rotatively driven by the vehicle wheel, the shaft being secured to the drive gear to rotate the drive gear in the same direction as the shaft is rotated.

4. An electric reverse cut-off switch as set forth in claim 3 wherein said means for electrically connecting said movable arm to said cam gear actuable switch includes a contact button and means for moving said electrically conductive shaft endwise to bring said contact button into engagement with said movable arm when said shaft is rotated in the first direction and disengaged from contact with said movable shaft when said shaft is rotated in the second direction.

5. An electric reverse cut-off switch as set forth in claim 4 wherein said means for moving said electrically conductive shaft endwise includes means for biasing said electrically conductive shaft endwise into engagement with said contact button when said shaft is rotated in the first direction and cam means for moving said shaft endwise out of engagement with said contact button when said shaft is rotated in the second direction.

6. An electric reverse cut-off switch as set forth in claim 5 wherein said cam means includes an apertured disc-shaped plate mounted on said electrically conductive shaft, said plate being formed with a spiral cam surface, a cam pin connected to and extending radially of said shaft, said cam pin being positioned to engage the spiral cam surface such that when said shaft is rotated in the second direction the cam pin will slide along the spiral cam surface and urge said shaft endwise out of engagement with said contact button.

7. An electric reverse cut-off switch as set forth in claim 6 including means for biasing said contact button toward said endwise movable electrically conductive shaft and means for holding said disc-shaped plate against rotary movement as said shaft rotates.

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[PX 12, p 1]

PLAINTIFF'S EXHIBIT 12

Aug. 5, 1969 F. A. PERRINO 3,459,204

ANTI-SKID DEVICE VALVES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 1

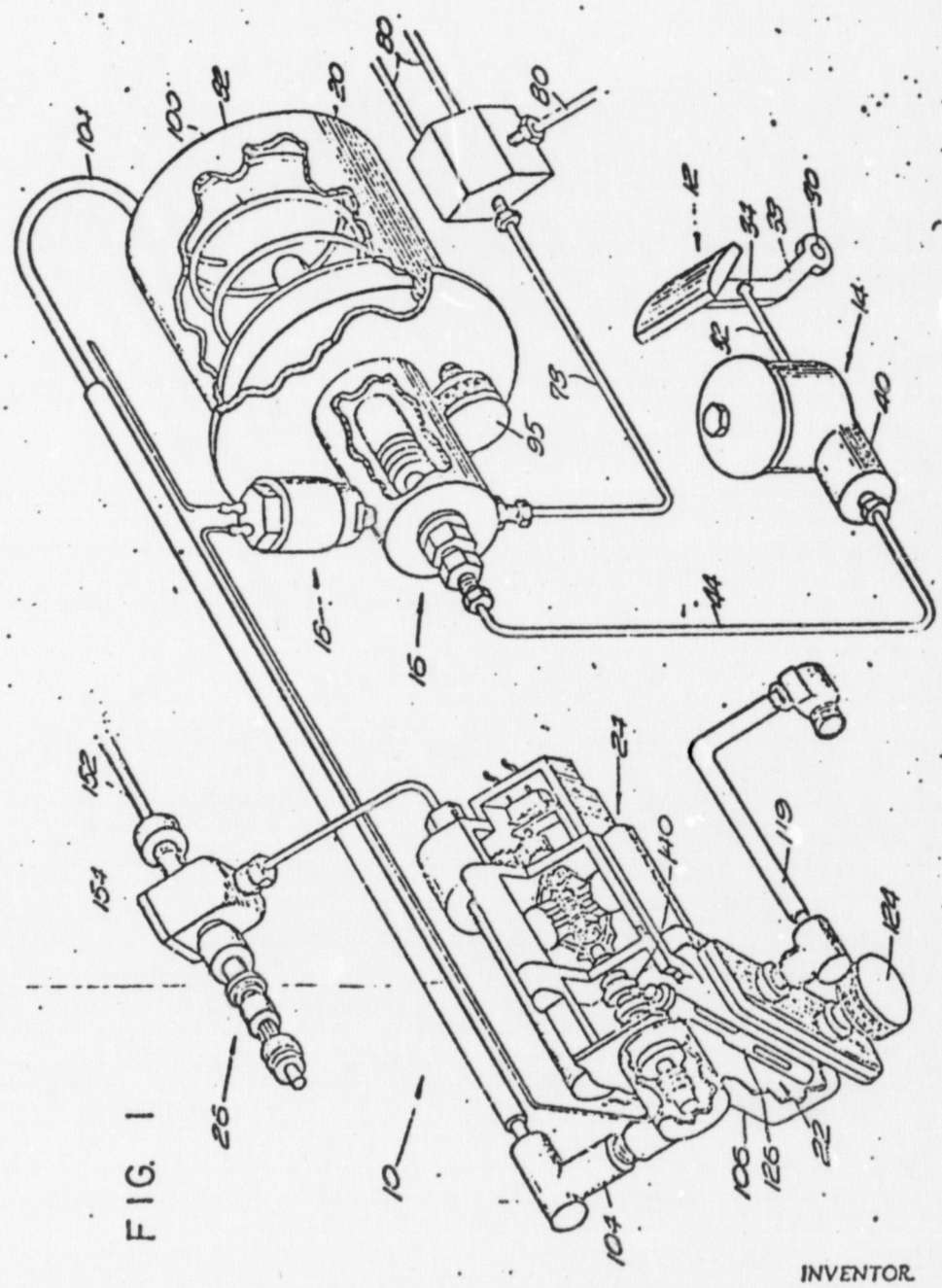


FIG. 1

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ANTI-SKID DEVICE VALVES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 2

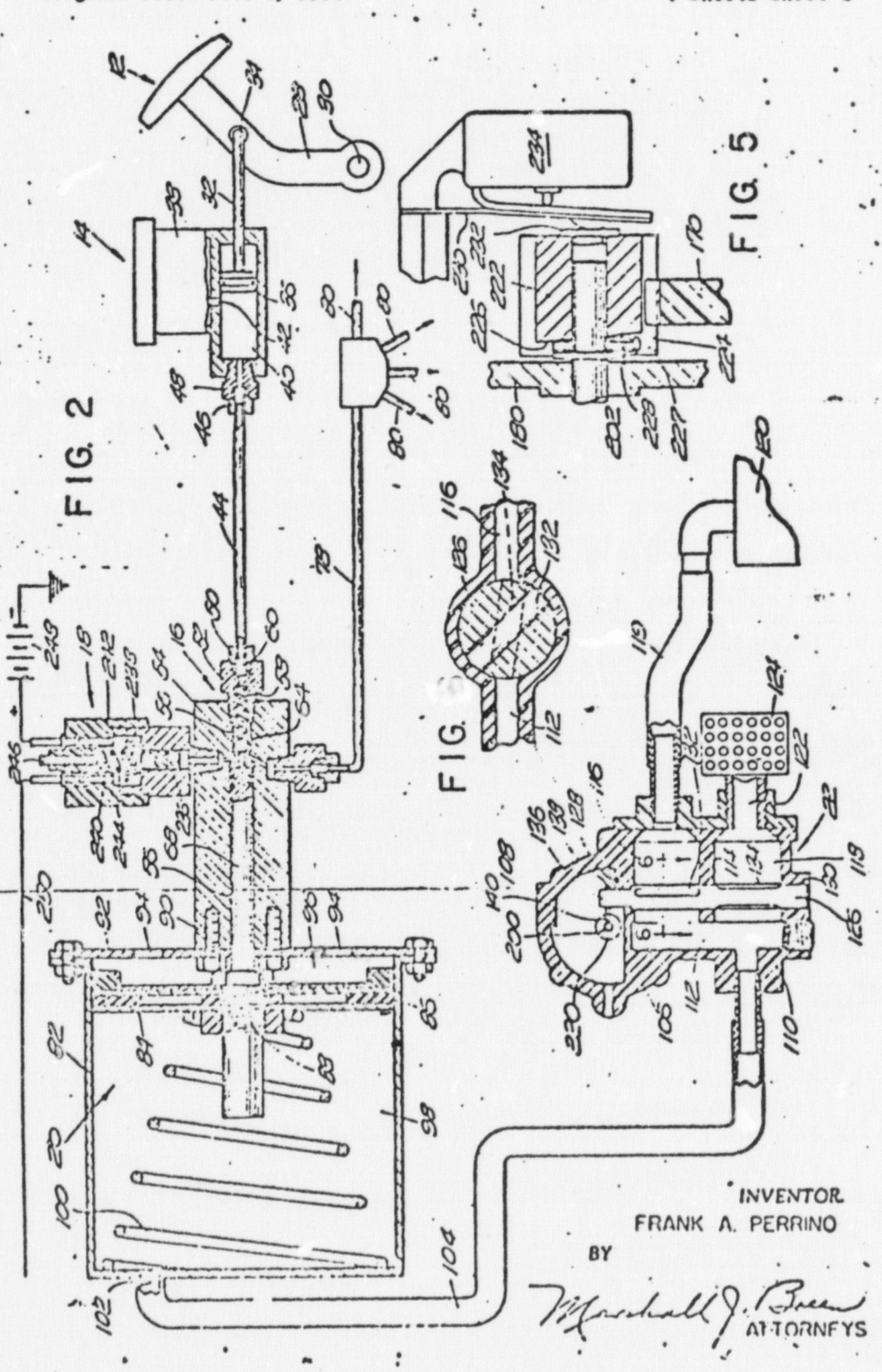
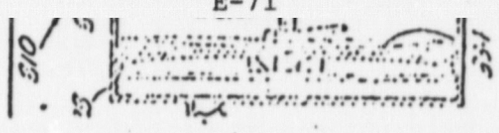


FIG. 2

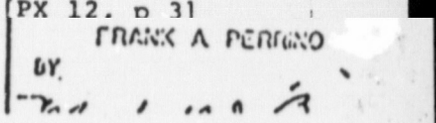
FIG. 5

FIG. 6

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[PX 12, p 3]



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[PX 12, p 3]

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ANTI-SKID DEVICE VALVES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 3

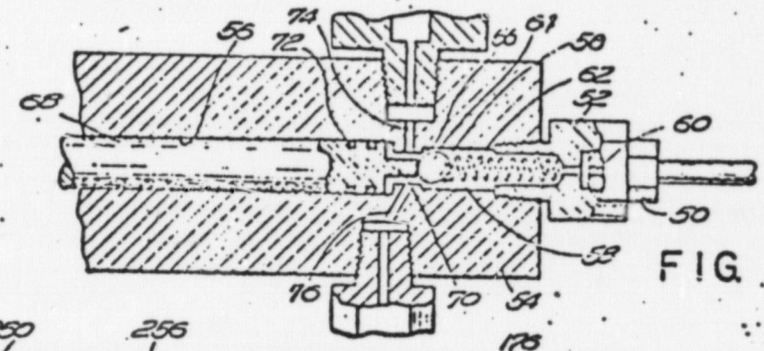
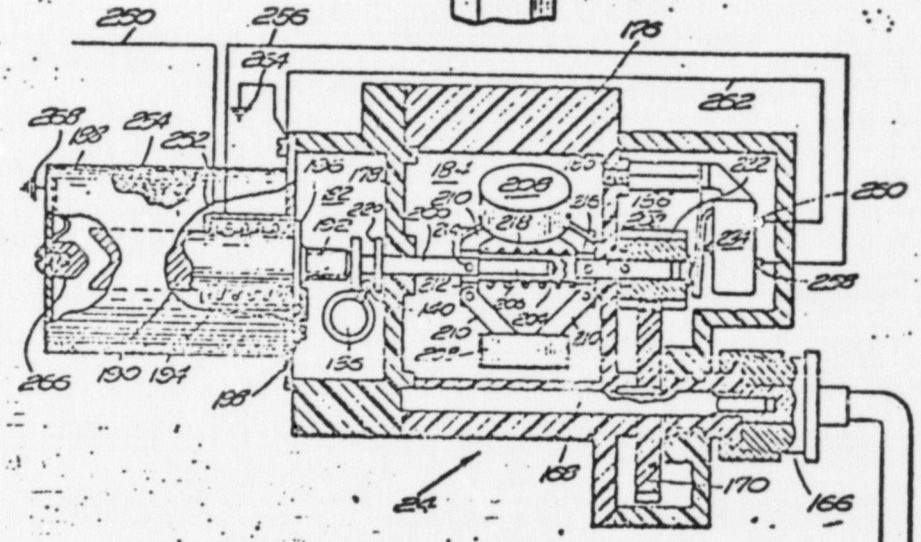


FIG. 3



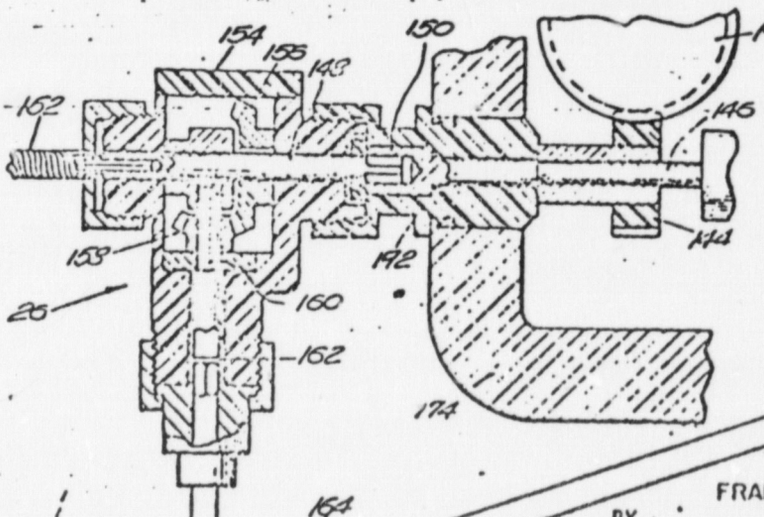




FIG. 4

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ANTI-SKID DEVICE VALVES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 4

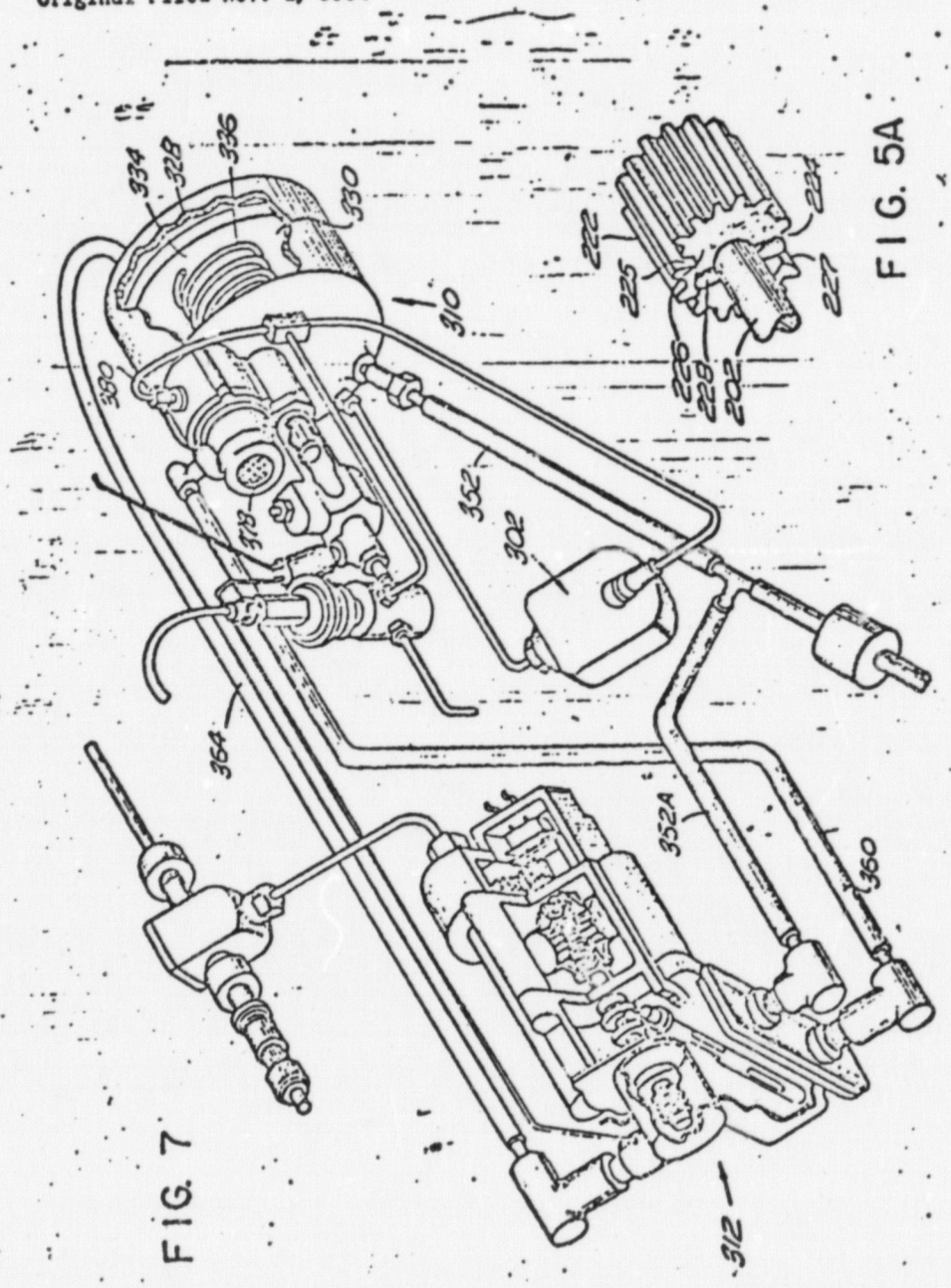


FIG. 5A

FIG. 7

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Original Filed Nov. 2, 1964 7 Sheets-Sheet 5

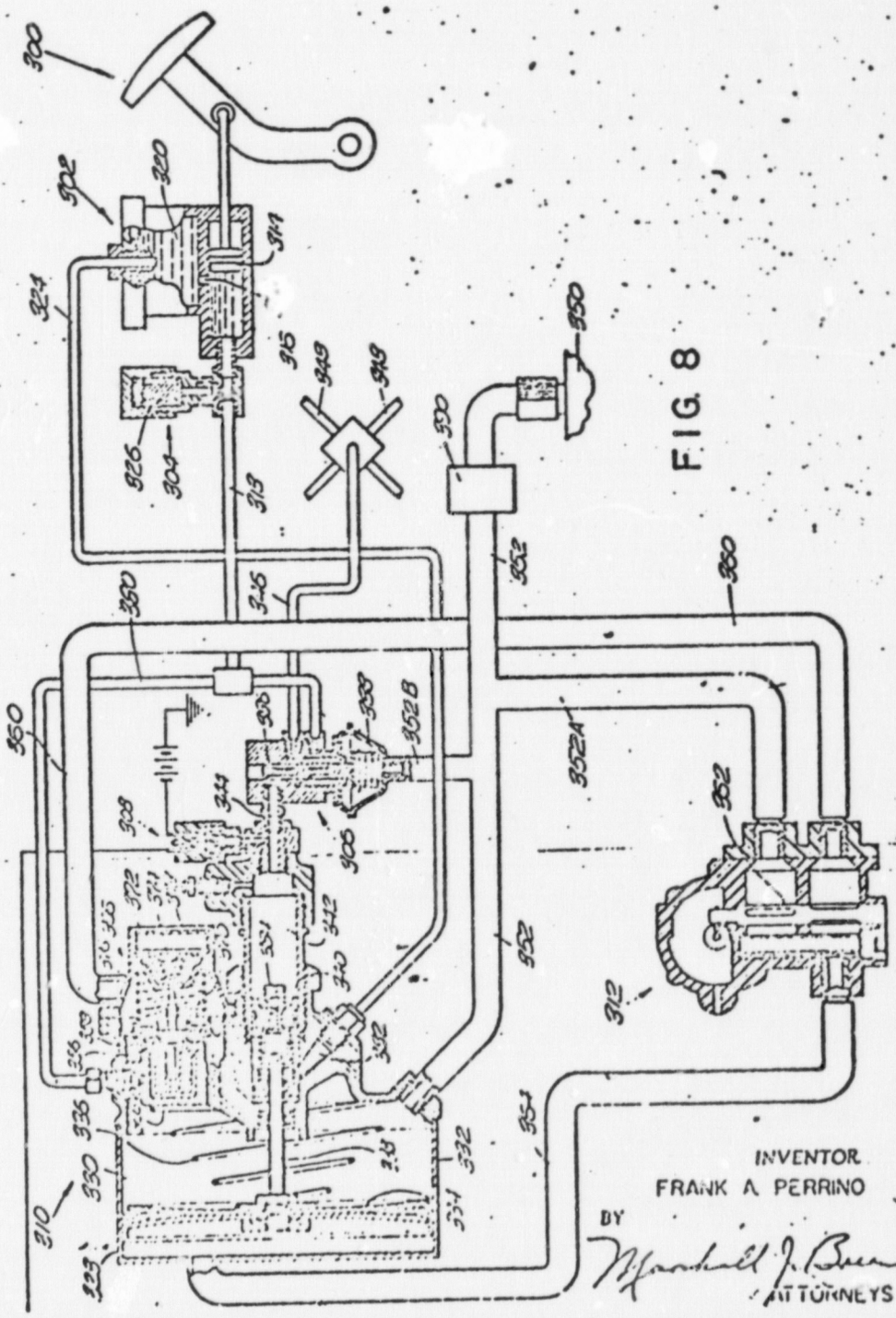


FIG. 8

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can act before this occurs, whereby normal control is

[PX 11, p 8]

[PX 12, p 6]

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ANTI-SKID DEVICE VALVES

Original Filed Nov. 2, 1964 7 Sheets-Sheet 6

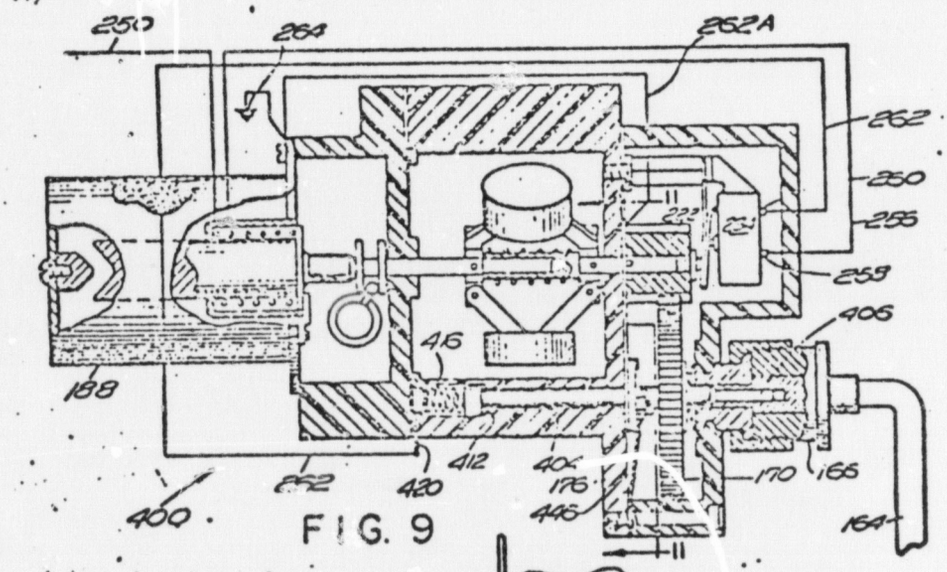


FIG. 9

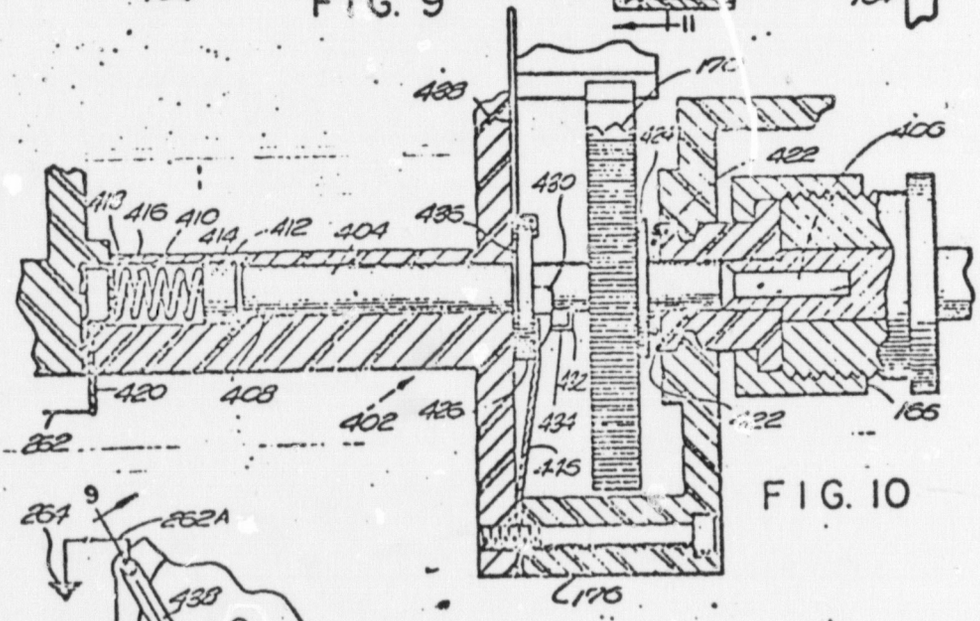


FIG. 10

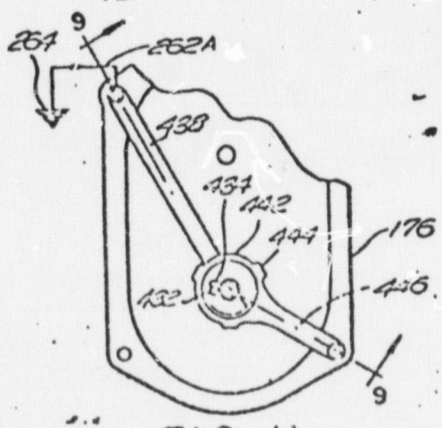


FIG. 11

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[PX 12, p 7]

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ANTI-SKID DEVICE VALVES

Original Filed Nov. 2, 1964 7 Sheets-Sheet

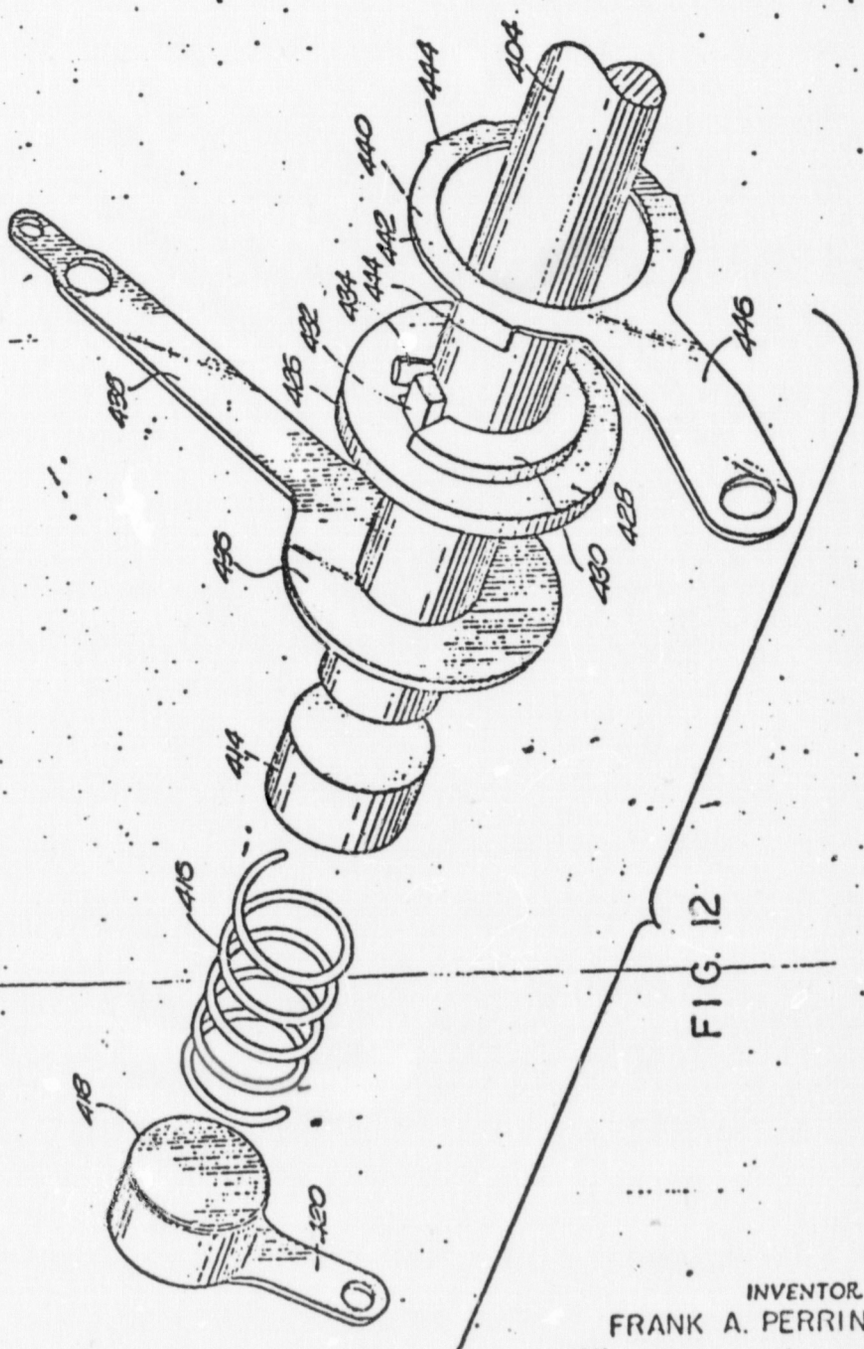


FIG. 12

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United States Patent Office

3,459,204

Patented Aug. 5, 1969

3,459,204
ANTI-SKID DEVICE VALVES

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Original application Nov. 2, 1964, Ser. No. 409,350, now Patent No. 3,325,226, dated June 13, 1967. Divided and this application Aug. 5, 1966, Ser. No. 594,291

Int. Cl. B60t 8/02

U.S. Cl. 137—38 6 Claims

ABSTRACT OF THE DISCLOSURE

A rotary valve formed with axially spaced openings that selectively connect two different pressure sources to a vehicle wheel brake operating actuator cylinder in an antiskid device.

This is a division of U.S. Patent No. 3,325,226, issued on June 13, 1967, and application Ser. No. 409,350, filed on Nov. 2, 1964, and entitled Anti-Skid Device Valves.

The present invention relates generally to braking systems for automotive vehicles and the like and is particularly concerned with the provision of novel and improved anti-skid means.

It is well known that a skid condition may be encountered by a moving vehicle where the brakes of the vehicle are applied so hard that the vehicle wheels tend to lock. Quite obviously, other factors affect the commencement and existence of a skid condition, such as the condition of the surface on which the vehicle is moving, the speed at which the vehicle is traveling, etc. However, it is possible for a vehicle to commence a skid even where the surface on which the vehicle is traveling is not wet or icy and even where the vehicle is traveling at moderate speeds, if the vehicle brakes are applied too hard. Since all control of a vehicle is lost while the vehicle is in a skid, thus increasing the danger of serious accident and personal injury, it is obviously highly desirable to prevent or reduce skidding while at the same time enabling the vehicle to come to a complete stop as quickly as possible.

It is therefore a primary object of the present invention to provide an improved braking system wherein operation of the brakes is under control of the vehicle operator until a skid condition commences to exist, at which time anti-skid means automatically take over complete control of the brakes and maintain said control until danger of skidding no longer exists, whereupon the brakes are returned to normal control by the operator.

Another object of my invention is the provision of efficient and sensitive control means for the various operations of the system, which have usefulness in other types of systems as well.

Another object of my invention is the provision of novel and improved anti-skid means equally adaptable to so-called power brakes, as well as nonpower brakes.

A further object of my invention is the provision of anti-skid means of the character described wherein the vacuum system of the vehicle is utilized, but wherein the loss of vacuum in no way interferes with control of the vehicle brakes by the operator.

A further object of my invention is the provision of anti-skid means that is readily and easily adaptable to existing vehicles.

Another object is the provision of a novel and improved reverse cut-off switch that automatically renders the anti-skid means inoperative when the vehicle moves in reverse.

Still another object of my invention is the provision of an anti-skid system for automotive vehicles and the like having means which are durable and effective in operation and which are feasible and practical both from an economic and manufacturing standpoint.

Other objects, features and advantages of my invention will become apparent as the description thereof proceeds when considered in connection with the accompanying illustrative drawings.

In the drawings which illustrate the best mode presently contemplated for carrying out the present invention:

FIG. 1 is a diagrammatic view, in perspective, of the assembly utilized in my nonpower unit, with portions being broken away for purposes of illustration;

FIG. 2 is an elevational view, in section, of a portion of the overall assembly illustrated in FIG. 1;

FIG. 3 is an enlarged fragmentary sectional elevation of a portion of the brake-actuating means illustrated in FIG. 1;

FIG. 4 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid unit and the drive means therefor;

FIG. 5 is an enlarged sectional elevation of the cam gear and solenoid-actuating switch which form a part of the anti-skid unit;

FIG. 5A is an enlarged fragmentary perspective view of the cam gear;

FIG. 6 is an enlarged section taken on line 6—6 of FIG. 2;

FIG. 7 is a diagrammatic view, in perspective, of my anti-skid means in combination with a power brake unit, portions being broken away for purposes of illustration;

FIG. 8 is an elevational view, in section, of the power brake assembly illustrated in FIG. 7;

FIG. 9 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid means and the reverse cut-off switch in combination therewith;

FIG. 10 is an enlarged fragmentary sectional elevation of the reverse cut-off switch per se;

FIG. 11 is a fragmentary section taken on line 11—11 of FIG. 9; and

FIG. 12 is an enlarged exploded perspective view of the reverse cut-off switch per se.

According to one aspect of my invention, a sensing means for sensing when a skid condition begins is caused to activate a wheel speed-sensitive device that takes over control of the brake pressure until the skid condition ends. Advantageously, the sensing means is an inertia-responsive means that senses a sudden change in the speed of rotation of a wheel at the commencement of skid conditions. This inertia-responsive device does not control the brake pressure itself by repeated action during the skid condition, as has been attempted by the prior art. Instead the control of the brake pressure is taken over completely by a governor driven dependently with the wheel, advantageously a mechanical governor. The governor then controls the brake pressure in a hunting or modulating manner, for an extended period, until such control of the brakes is ended by a reset means that responds to slowing of the vehicle to a predetermined speed. Preferably, the reset means is a brake-pressure-sensitive device which senses reduction of brake pressure to a predetermined level which the governor can produce only when it slows to the predetermined speed.

This separation of the initiating and controlling functions is an extremely important feature. The initiating device must necessarily have a restraint to prevent operation when the vehicle merely decelerates normally. I have realized that such a restraint prevents the device from being able to effectively regulate the brake pressure due to leak

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of sensitivity. The governor, after it takes over control, is free of such restraint and therefore can have much greater sensitivity with the result that the brake pressure can be much more accurately regulated.

The reset device is also of critical importance because at low speeds, e.g. below 10 m.p.h., the governor forces can become so diminished that the governor can behave entirely as if the vehicle were skidding, though not, thereby totally relieving the brake pressure. The reset device can act before this occurs, whereby normal control is regained.

A further aspect of my invention concerns the realization that a predetermined level of brake pressure, e.g. the minimum brake pressure at which skidding of the vehicle will occur on ice, generally between 100–150 p.s.i. for conventional automobile, can be utilized to determine the setting for the brake pressure-sensitive reset device. Thereby, only at the time when skidding is no longer possible will control be taken away from the anti-skid unit to return the brake system to normal operation. This pressure-sensitive device, particularly when taking the form of a brake fluid pressure-sensitive switch, is useful with anti-skid systems generally, though of particular usefulness in the system described above.

Advantageously, the brake pressure-sensitive switch operates an electrical system, preferably a solenoid, to restore normal brake operation, and it can be employed to bring the anti-skid unit into readiness whenever the operator applies his brakes, as well as to stop the anti-skid unit from acting when skid conditions have ended.

Another aspect of the invention concerns the control of the governor. In normal operation of the brakes, the governor is prevented from influencing the brakes, but once activated, e.g. by the inertia-sensitive device, it is made continually to act, even when the vehicle wheel speeds up after the brakes have been initially relieved. A controlling or restraining member, preferably a solenoid, is employed to achieve this operation, with the controlling member held steadily from interfering with the governor until the reset device operates. In the case where a rotary weight mechanical governor is employed, a plunger can normally restrain the governor and a solenoid, initially under the control of the inertia-responsive device, can pull back the plunger to free the governor, whereupon the plunger can operate a switch to maintain the solenoid energized, and at the end of skid conditions the reset device can de-energize the solenoid to once again restrain the governor.

Another aspect of the invention is a fluid valving system that responds to the governor to increase and decrease brake pressure. Advantageously, the valve, discharging to a brake control conduit, is moved by the governor alternately between two positions to connect two passages for fluid at different pressures to the conduit. This causes the brake pressure to rapidly vary in a modulating or hunting manner, to keep the brake pressure at maximum values while preventing skidding. The mean about which the pressure modulates changes as the slipperiness of the road changes, due to the effect such road changes have upon the governor.

For use in a closed-circuit hydraulic system, the direct control of the brakes is achieved, according to the invention, by an isolation valve that continually seals the wheel brake from the control of the operator, and a variable volume chamber, e.g. piston and cylinder, for relieving and reapplying brake pressure to the wheel, the piston modulating about a mean determined by the modulating pressure conditions of the brake control conduit. Advantageously, the piston, at the position where it allows full pressure, can operate the isolation valve. Then the piston being spring loaded to that position can also perform a pressure-limiting function, limiting brake pressure to e.g. 1,000 p.s.i., though the operator is capable of applying twice that much. In normal operation the brakes function as well at 1,000 p.s.i. as above, and by so limiting the brake pressure, less is pressure to be relieved when a skid occurs, which enables faster action and allows the relief chamber to be smaller.

In other brake systems the brake control conduit can control a power unit, or in the case of air brakes, it can directly control the brake actuators associated with the wheels.

Furthermore, according to the invention, I have provided advantageous and improved constructions of the various operating components and sub-systems which can best be described with reference to the drawings.

Referring now to the drawings, and more particularly to FIGS. 1-6 thereof, a closed hydraulic circuit braking system employing my novel and improved anti-skid means in combination with a so-called nonpower brake will be described. The over-all system, generally designated as 10, comprises brake pedal 12, master cylinder 14, brake-relieving device 16, hydraulic pressure switch 18, relief actuator means 20, conduit valve means 22, skid-sensing and control means 24, and anti-skid unit drive means 26.

Referring to FIGS. 1 and 2, the brake pedal 12 is of conventional construction and comprises bell crank 28 pivotally connected to a suitable support on the vehicle (not shown) at 30 whereupon depression of pedal 12 causes bell crank 28 to rotate in a counterclockwise direction around pivot point 30, thus actuating connecting rod 32 pivotally connected to bell crank 28, as at 34. Rod 32 carries at its other extremity a piston 36 which functions to pump oil or other suitable fluid from master cylinder 14, all in a well-known manner. Specifically, master cylinder 14 has an upper reservoir 38 which communicates with lower chamber 40 by means of opening 42 whereupon a full supply of oil is always present in the chamber 40 to be pumped by actuation of pedal 12 and piston 36.

Fluid pumped from master cylinder 14 passes through a conduit 44 connected at one of its extremities by suitable couplings 46, 48 to chamber 40 of master cylinder 14 and at its other extremity by a suitable coupling 50 to the brake-relieving device generally designated at 16, which incorporates an isolation valve. More specifically, coupling 50 is connected to a second coupling 52 which is threadedly attached to block 54, said block having a bore 56 extending longitudinally therethrough into communication with aligned longitudinally extending bores 58 and 60 in couplings 52 and 50, respectively.

As will be seen most clearly in FIGS. 2 and 3, coupling 52 is provided with an internal shoulder for receiving positioning spring 62 which positions isolation valve ball 64 adjacent valve seat 66, tending to close with slight force the bore or passageway 56. During normal operation of the vehicle, ball 64 is maintained in its unseated position, as illustrated in FIGS. 2 and 3, against the action of spring 62, by means of spring-loaded actuator piston 68 having a reduced extension 70 in engagement with ball 64. Suitable O-rings or the like 72 are provided to insure a tight sliding fit between piston 68 and bore 56 whereby said piston effectively blocks further flow of fluid through bore 56.

Block 54 is provided with a pair of oppositely disposed passageways 74 and 76, it being noted that said passageways communicate with bore 56 just behind valve seat 66, whereupon when ball 64 is in its closed position in engagement with valve seat 66, fluid in conduit 44 from the master cylinder will be blocked before reaching the passageway 74 and 76. Passageway 74 communicates with hydraulic pressure reset switch 18 for reasons hereinafter to be described, while passageway 76 communicates by means of suitable couplings with conduit 78, which in turn communicates with the hydraulic brake lines 80 leading to the wheel hydraulic actuator cylinders (not shown) of the vehicle to actuate the brakes in well known fashion. Thus it will be seen that with the parts in the positions illustrated in FIGS. 2 and 3, and particularly with

piston 68 in the position illustrated, brake pedal 12 controls actuation of the vehicle brakes through a direct line. This is the normal operating condition of the vehicle.

The actuator for the relieving device 16 generally designated at 20 (FIGS. 1, 2) controls the movement of piston 68 and hence ball 64. This actuator comprises a housing or cylinder 82 having a large piston 84 slidably mounted therein, said piston having suitable seals 86 at its outer periphery to insure that the piston 84 will make a tight sliding fit within cylinder 82 whereby no leakage will exist around the piston. Piston 84 is threadedly connected, as at 88, to the aforementioned small hydraulic piston 68, whereupon movement of large piston 84 causes corresponding movement of piston 68. Housing 82 is secured to block 54 by any suitable means, such as screws 90, and the adjacent end wall 92 of the housing 82 is in free communication with atmosphere, as by ports 94. Suitable air filters, such as shown at 95 in FIG. 1, may be associated with the ports 94. It will thus be seen that the chamber 96 located on one side of large piston 84 is always at atmospheric pressure, while chamber 98, located on the opposite side of piston 84, may either be at atmospheric pressure or it may be vacuum charged, as will hereinafter be described in more detail. When the chambers 96 and 98 are both at atmospheric pressure, spring 100 resiliently urges piston 84 to the position illustrated in FIG. 2, it being seen that the piston 84 has moved to the right as far as it can go. In this position, the piston 68 has also been axially moved to the position illustrated, in which position extension 70 (FIG. 3) has engaged ball 64 to unseat it from valve seat 66. Thus, where chambers 96 and 98 are both at atmospheric pressure, the vehicle brakes are under the complete control of the brake pedal 12. However, when chamber 98 is vacuum charged, the pressure differential between chambers 96 and 98 is sufficient to overcome spring 100, whereupon piston 84 will move to the left, viewing FIG. 2, thus causing corresponding movement to the left of piston 68. This in turn enables ball 64 to seat against valve seat 66, by means of spring 62, thus blocking the bore 56 and preventing actuation of the brakes by means of brake pedal 12. As pistons 84 and 68 move further to the left due to the presence of a vacuum or even a partial vacuum in chamber 98, hydraulic pressure in lines 78 and 80 and in the spring-loaded brake actuators associated with the vehicle wheels will be relieved, due to the fact that the fluid is now free to back up into that portion of bore 56 that has been vacated by movement of piston 68 to the left. It will therefore be seen that by introducing a vacuum or partial vacuum to chamber 98, fluid pressure in lines 78 and 80 will automatically be relieved, thereby relieving the brakes. By the same token, as soon as atmosphere is again introduced to chamber 98 to equalize the pressure in chamber 96, the brakes can once again be fully actuated by the pedal 12. It will be apparent that the amount of vacuum introduced to chamber 98 will determine the amount of reduction in brake pressure. More specifically, the greater the vacuum that is introduced to chamber 98, the greater will be the movement of piston 84 to the left (viewing FIG. 2) and at the same time there will be greater movement of piston 68 to the left, thus vacating more of bore 56 and providing greater relief of the hydraulic brake pressure. Similarly, given a certain amount of relief due to a certain amount of vacuum, then decrease in the amount of vacuum will increase the hydraulic brake pressure due to movement of the pistons to the right. The ability of my system to provide varying degrees of relief and restoration of brake pressure while the brake pedal remains isolated from the system is an important and advantageous feature.

It will be noted that during normal operation of the system, as heretofore described, i.e., where atmospheric pressure exists in both chambers 96 and 98, the piston 68 will function as a pressure limitor in the system. More specifically, since the piston 68 is resiliently acted in the

position illustrated in FIG. 2 by means of spring 100, it will back off when excess pressure is applied by foot pedal 12. In practice it has been found that in a conventional automobile pressure above 1,000 p.s.i. or at least above 1,500 p.s.i., depending on the type of vehicle, is unneeded for normal brake operation and hence springs 62 and 100 are preferably calibrated so as to enable ball 64 to seat when pressure in excess of that needed is introduced, thus preventing introduction of excessive pressure to the brakes. It will be obvious that the springs may be calibrated so as to provide a limit at any selected pressure.

The means for introducing either atmospheric pressure or vacuum to chamber 98 and for modulating the degree of vacuum therein will now be described. Referring to FIG. 2, it will be seen that cylinder housing 82 is provided with an opening 102 in communication with chamber 98, which opening communicates with a brake control conduit 104 that in turn connects with the conduit valve, generally designated at 22. More specifically, there is provided a housing 106 having an upper portion 108 which houses the skid-sensing and control means 24, and a lower portion 110 which houses the valve 22. The lower housing 110 comprises an inner chamber 112 having a partition 114 defining a vacuum passage 116 and an atmosphere passage 118. More specifically, vacuum passage 116 is in communication, by means of passage 119, with the intake manifold 120 of the vehicle. Thus, suction is constantly being applied to the passage 116. The passage 118, on the other hand, is in communication with atmosphere through port 122, to which a suitable air filter 124 may be attached, if desired. A rotary valve 126 is mounted for rotation in housing 106, it being noted that valve 126 is an elongated rod that extends vertically through upper partition 128, the aforesaid partition 114, and is journaled in bottom wall 130, the partition and wall being integral parts of a unique molded housing of antifriction plastic. The valve 126 is provided with a pair of passageways or bores 132 and 134 extending diametrically therethrough, said bores being vertically spaced from each other, and having elongated cross-sections in the axial direction, longer than the diameter of the rod. Bore 132 is associated with passage 116 and bore 134 with passage 118. The bores are angularly disposed with respect to each other, as shown most clearly in FIG. 6. Thus, with the rotary valve 126 in the position illustrated in FIGS. 2 and 6, bore 134 is positioned so as to allow free passage of atmosphere from passages 122 and 118, bore 134 and chamber 112 to brake control conduit 104, whereby atmosphere will flow into chamber 98. At the same time, due to the angular displacement of bore 132, passage 116 will be closed by valve 126, thus preventing suction from manifold 120 and passage 119 from coming into contact with chamber 112 and brake control conduit 104. It will be understood that the only access from passage 116 and 118 to chamber 112 is through valve 126 and more specifically, the bores 132, 134 thereof. Should valve 126 be rotated by means hereinafter to be described, it will be seen that communication between passage 118 and chamber 112 will eventually be blocked, thus interrupting the flow of atmosphere to brake control conduit 104, while at the same time passage 116 will be in communication with chamber 112, whereupon a suction will be applied through brake control conduit 104 to chamber 98.

It is important to note that use of a rotary valve such as valve 126 is extremely advantageous where passage of a vacuum and atmospheric pressure, or more generally two different pressures, is being controlled due to the fact that the pressure differential offers a minimum of resistance to the turning movement of the small diameter valve. This is contrasted to a reciprocal or poppet type valve wherein movement of the valve is much more resisted by pressure differential.

As hereinbefore explained and described, introduction of a vacuum or partial vacuum to chamber 98 automati-

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cally tends to release brake pressure on the vehicle wheels. It has been found in practice that an angular displacement between bores 132 and 134 of approximately 37.5° gives best results in the operation of my system, and it will be understood that rotary valve 126 may not always be positioned so as to completely open or close the bores 132 and 134. Thus, different degrees of movement of valve 126 allow different brake fluid pressures to be exerted on the vehicle wheels, thus providing the proper braking pressures for different road conditions. In instances where the pressure on the brake actuator is the same at the beginning of a skid, (lesser initial uresure requires less relieving), the amount of relief of the brake pressure depends on the amount of travel of piston 84 to the left in chamber 98 which in turn is dependent on the amount of vacuum introduced to the chamber. Thus, where the vehicle is braked and skids on a dry surface which requires a greater brake pressure to create a skid condition, lesser movement of piston 84 to the left, and hence less vacuum, will be required to relieve the brake pressure sufficiently to prevent the vehicle wheels from locking, this pressure being referred to as "release" pressure. Conversely, where the vehicle is braked or a slippery surface such as ice, a lesser brake pressure is required to create a skid condition. Hence the valve 126 must open suction passage 116 to the brake control conduit 104 for a longer initial period to provide greater vacuum to chamber 98 in order to move piston 84 sufficiently to the left so that sufficient relief of the brake pressure will be effected to prevent the wheels from locking.

In accordance with the invention after release pressure is attained, with corresponding movement of the piston then the governor causes the degree of vacuum in conduit 104 to modulate, which modulates the brake pressure through a narrow range of intermediate pressures above and below the release pressure, without ever restoring the normal, higher brake pressure until the skid condition ends.

To thus operate the valve, the upper extremity of valve 126 (FIG. 2) is provided with a reduced portion 136 that extends upwardly through partition 128, said portion 136 having a pin 138 extending therethrough to which is attached a circular projection 140, it being seen that as a longitudinal thrust is exerted against projection 140, the valve 126 will be caused to rotate.

The means for controlling movement of valve 126, namely, the skid-sensing and governor control means and the drive means therefor, will now be described. Referring to FIG. 4, skid-sensing and governor control means is designated generally at 24, and the drive means therefor is designated generally at 26. Referring first to the latter, the vehicle drive shaft is shown at 142, it being understood that the drive shaft is directly connected to the wheels of the vehicle (not shown) whereupon when the vehicle wheels lock, the drive shaft also locks. The drive shaft drivingly engages speedometer gear 144, which in turn is fixed to shaft 146, which in turn is drivingly connected to shaft 148 by means of a coupling shown at 150, said shaft 148 connecting with speedometer cable 152. Gear box 154 through which shaft 148 passes and in which the shaft 148 is suitably journaled, houses drivingly engaged bevel gears 156, 158, it being understood that bevel gear 156 is keyed to shaft 148 and bevel gear 158 is keyed to shaft 160, whereupon rotation of shaft 148, transmitted thereto by the vehicle drive shaft 142, is transmitted to shaft 160. Shaft 160, in turn, is drivingly interconnected, as at 162, to flexible shaft 164, said flexible shaft being connected, as at 165, to the unit 24, and said flexible shaft being drivingly interconnected with a shaft 168 which has fixed thereto a drive gear 170. It will therefore be seen that the gear 170 is drivingly interconnected with the vehicle drive shaft 142, which in turn is connected to the vehicle wheels. Thus, rotation of the vehicle wheels will cause corresponding rotation of gear 170, and, conversely,

sudden slowing of the vehicle wheels, such as would occur where the vehicle approaches a skid condition by the operator applying more brake pressure than is required for a particular road surface, will cause not only the main drive shaft 142 to suddenly slow, but also drive gear 170, Gear box 154, and adaptor 172, by means of which the gear box is connected to the engine housing 174, comprise an assembly which may be easily and readily adapted to a conventional vehicle for driving the sensing and governor control means now to be described.

The unit 24 (FIGS. 1, 4) comprises a molded plastic housing 176 having a pair of spaced partitions 178, 180 therein defining chambers 182, 184 and 186. A solenoid 188 is secured to housing 176 adjacent chamber 182 by any suitable means, the solenoid 188 having a plunger 190 with a reduced extension 192 resiliently urged into chamber 182 by means of spring 194. When solenoid 188 is not energized, spring 194 bears against flange 196 affixed to plunger 190 to resiliently urge extension 192 into chamber 182. Suitable stop means 198 mounted on the interior of chamber 182 engaged flange 195 to limit resilient axial movement of plunger 190 and extension 192 at a predetermined point, for reasons hereinafter to be explained more fully. Journaled in the partitions 178 and 180 are a pair of aligned shafts 200 and 202, respectively. Shaft 202 is provided with an enlarged portion 204 having an internal bore for slidably receiving in telescoping relation a reduced extension 206 of shaft 200. Attached to shafts 200 and 202 is a governor assembly comprising a plurality of weights 208, each pivotally connected to a pair of links 210, which links in turn are pivotally connected to arms 212. The arms 212 located on one side of the weights 208 are affixed to shaft 200 as at 214, while the arms on the other side of the weights are affixed to shaft 202 as at 216. As hereinbefore pointed out, shaft 202 is journaled for rotation in partition 180 but is incapable of longitudinal movement. Shaft 200, on the other hand, is drivingly engaged with shaft 202 so as to rotate therewith, but said shaft 200 is axially slidable with respect to the said shaft 202. Since the governor assembly is affixed on one side to shaft 202 and on the other side to shaft 200, it will be seen that the speed of rotation of the shafts will determine their longitudinal position relative to each other. Spring 218 resiliently urges the governor assembly to its collapsed position wherein shafts 200 and 202 are axially extended with respect to each other, but as the rotational speed of shafts 200 and 202 increases, centrifugal force urges the weights 208 outwardly, thus moving shaft 200 toward the right, viewing FIG. 4, against the action of spring 218.

Fixed to shaft 200 at the extremity thereof located in chamber 182 is a spool member 220, which spool member is in engagement with projection 140 carried by the reduced upper extension 136 of rotary valve 126. Thus, axial movement of shaft 200 causes like movement of the spool 220 which in turn carries therewith projection 140 captured between the opposed walls of the spool to thereby impart rotary movement to valve 126.

The means for imparting rotational movement to shafts 200 and 202 and the governor assembly associated therewith comprises a cam gear 222 (FIGS. 4, 5 and 5A), which cam gear is part of the drive train, in driven engagement with gear 170. The cam gear 222 is mounted on shaft 202 and is located in chamber 186. The cam gear is slidably mounted on shaft 202 and is in driving engagement therewith by means now to be described. A recess 224 having inclined or helical surface 226 is provided at the end of the cam gear adjacent partition 180, and a projection or pin 228, carried by shaft 202, extends into said recess in engagement with the cam surface 226. At either end of the recess is a stop surface, at the deeper end stop surface 227, adapted to engage pin 228 for accelerating the governor and at the shallower end stop surface 225 for engaging pin 228 to decelerate the governor. A leaf spring 230 mounted to the housing 176 by any suit-

through opposite [illegible] through rotary valve 252 of the anti-skid unit 312, through conduit 304 and tee 50 to chamber 328. The control unit generally shown at 316 which forms a part of power brake unit 312 is of conventional form and com-

conduit 309 becomes [illegible] conduit 305A becomes open to allow suction from manifold 334 to be introduced through conduit 254 to chamber 328, whereupon spring 336 will once again take over to release the

able means engages a button 232 which is press fitted into the central bore of cam gear 222 to normally urge the latter axially toward partition 180, holding pin 228 against accelerating stop surface 227. However, when rotation of drive gear 170 and cam gear 222 is suddenly slowed, as by sudden slowing of the vehicle wheels upon the presence of a skid condition, the rotational inertia of shaft 202, especially due to the governor weights 208, causes shaft 202 to turn ahead of gear 222, causing pin 228 to leave stop surface 227 and move to stop surface 225. This slides gear 222 on shaft 202 in a direction away from partition 180 to activate governor control.

It is important to note that solenoid spring 194 is stronger than governor spring 218, whereupon when solenoid 188 is not energized, plunger extension 192 bears against spool 220 to restrain same in the position illustrated in FIG. 4 wherein the rotary valve 126 is in the position illustrated in FIG. 2, namely, the position where atmosphere is free to pass through the valve to conduit 104 and then to chamber 98 and wherein at the same time suction from the intake manifold is blocked from communication with conduit 104. This is the normal operating condition of the system wherein the balanced pressure on opposite sides of actuator piston 84 enables spring 100 to position piston 84 and piston 68 as illustrated in FIG. 2. In this position, ball valve 64 is unseated to allow fluid pressure to pass directly to the brake line whereupon operation of the brakes is under direct control of brake pedal 12. This condition will remain so long as solenoid 188 (FIG. 4) is not energized. Energization of solenoid 188 is accomplished in the following manner. As previously explained, sudden slowing of gear 170 due to sudden slowing of the vehicle wheels as a result of a skid condition causes cam gear 222 (FIGS. 4, 5, 5A) to move axially on shaft 202 away from partition 180 against the action of spring 230. This movement is sufficient to cause actuation of a switch 234 suitably mounted in chamber 186. At the same time, the initial movement of the brake pedal and the resultant increase in fluid pressure in the system, causes hydraulic switch 18 (FIG. 2) to become closed. Hydraulic switch 18 is fully disclosed an described in copending U.S. patent application Ser. No. 203,075, filed June 18, 1962. Basically, switch 18 comprises piston 236 which has connected thereto at its upper end a metallic contact cup 238. Cup 238 is adapted to bridge contacts 240 and 242 but is normally maintained spaced therefrom by spring 244. An adjusting screw 246 is provided by means of which the tension of spring 244 may be varied, thus enabling the switch 18 to be preadjusted whereby a predetermined fluid pressure, e.g. 120–125 p.s.i., will cause piston 236 to move against the action of spring 244 sufficiently to bridge the contacts 240, 242. Actuation of switch 18 in the aforedescribed manner enables current to flow from battery 248 through contact 242, contact cup 238, and contact 240 to line 250 to the solenoid coil positive terminal 252 (FIG. 4). The voltage is applied through the solenoid coil 254 and then through line 256 to terminal 258 on switch 234. When switch 234 is open, the circuit to ground is not complete, and solenoid 188 will not be energized. However, upon actuation of switch 234 in the manner aforedescribed, the circuit to ground is completed through terminal 260, line 262 and ground connection 264. It is important to note that the circuit is such that although it is essential for hydraulic pressure switch 18 to be closed in order for solenoid 188 to be energized, closing of switch 18 alone is not sufficient, but rather switch 234 must also be actuated in order to complete the circuit. This is important because under normal operating conditions of the system sufficient pressure will normally be introduced to switch 18 by the foot pedal to actuate same, but it is essential that solenoid 188 not be energized under such normal conditions.

Energization of solenoid 188 accomplished by rightward movement of the cam gear (FIG. 5) closing switch 234, causes plunger 190, 192 to be retracted against the action of spring 194, thus freeing the shaft 200 and spool 220 for axial movement under the influence of the governor assembly. Therefore, the cam gear operating in response to the inertia of the governor and the solenoid operated plunger together can be regarded as an inertia-responsive means for initiating the action of the governor.

As soon as the solenoid plunger has been retracted, due to the sudden slowing at the commencement of a skid condition, the governor assembly takes over control and, as it is stopped by stop surface 225 of the cam gear, spring 218 will cause the governor assembly to commence to collapse, thus moving shaft 200 and spool 220 axially to the left, when viewing FIG. 4. This movement of spool 220 causes corresponding rotary movement of valve 126, thus shutting off the passage of atmosphere to brake control conduit 104 and simultaneously effecting communication between vacuum passage 119 and conduit 104. This results in suction being introduced to chamber 98 which in turn overcomes the action of spring 100 to cause pistons 84 and 68 to move to the left, viewing FIG. 2. This movement of piston 68 enables ball valve 64 to close due to the action of spring 62, thus preventing the introduction of further fluid pressure to the system via brake pedal 12. Further leftward movement of piston 68 enables the fluid in lines 78 and 80 to back up, thereby relieving the pressure until the vehicle brakes begin turning, the duration that the suction is applied, and hence the degree of vacuum and the amount of pressure relief depending on the slipperiness of the road. As soon as the brake release pressure is achieved, the vehicle wheels commence to rotate, whereby drive shaft 142 through the aforedescribed gearing once again causes gears 170 and 222 to rotate, it being apparent that this renewed rotation of shafts 200 and 202 will cause the governor assembly to once again turn, and due to the centrifugal force created, to retract shaft 200 and spool 220 to once again rotate valve 126 to the positions illustrated in FIGS. 2 and 4, where once again atmosphere is connected to chamber 98, thus enabling brake pressure to be restored.

But, it is very important for my present invention that the solenoid is not de-energized and the governor returned to the restraint of the inertia-responsive device. Rather, the governor remains free to act. It leaves the valve 126 open to atmosphere 118 for an instant until the degree of vacuum is lessened enough to cause slight rightward movement of the piston 84 to cause the brake pressure to slightly exceed release pressure and a skid again commences. Again the governor is slowed by the drive train and again it opens the valve 126 to suction 116, and this cycle repeats again and again in an extremely rapid bunting or modulating action with the gas pressure in the conduit 104 never reaching either atmospheric or engine vacuum level, but rather modulates at intermediate levels. In dependent fashion, the hydraulic brake pressure modulates narrowly about the release pressure as a mean, without returning to normal pressure. When the skid condition ends, the governor opens the valve continually to atmosphere and pistons 84 and 86 move fully to the right to once again unseat insulation valve 64 and restore normal brake operation. The faster the vehicle travels, the faster does this action occur, but the action is sufficiently fast down to speeds on the order of 10 m.p.h. and even lower, to rapidly pump and release the brakes. It is another very important aspect of the invention that the governor relinquishes control of the brake pressure before the vehicle stops.

In order to effectively carry out this modulating action, I have found it to be critical, in the preferred construction, that the solenoid 188 remain energized or that the plunger in some other way remains retracted until the vehicle slows to a predeterminted speed, e.g. 5 or 10 m.p.h., depending on the vehicle. Since closing of switch 234 to complete the circuit to ground that energizes solenoid 188 is only momentary, holding means are provided for main-

by means of spring 416. The opposite end of spring 416 resiliently engages a contact cup 413 having a terminal with line 266, completes the circuit to solenoid 188 it

[PX 12, p 13]

taining the solenoid energized until hydraulic pressure switch 18 opens, it being understood that switch 18 is preset to open only when fluid pressure in the system falls below the minimum value possible to achieve a skid condition, e.g. set to 85–90 p.s.i. produced by the governor at 5 or 10 m.p.h.

The aforesaid holding means for the solenoid comprise a terminal 266 having a connection to ground as at 268, which terminal is engaged by plunger 190 when the latter is retracted in order to complete the circuit to ground independent of switch 234. Thus, although closing of switches 18 and 234 is essential in order to energize solenoid 188, once the solenoid has been energized and its plunger retracted, switch 234 may be opened without de-energizing the solenoid, and the solenoid will remain energized until switch 18 opens.

As hereinbefore stated, it will be obvious that if switch 18 were not utilized, then there would be no means present for insuring de-energization of the solenoid once the condition has been reached where a skid is no longer possible, it being obvious that if the solenoid were still energised at such low speeds, the vehicle would be without effective brakes, since the governor assembly would not exert sufficient centrifugal force to overcome the action of spring 218 and move shaft 200 and spool 220 to the rightward position (FIG. 4) wherein valve 126 enables the brakes to be controlled by pedal 12.

It is possible for the reset device tot take some other form, such as a pendulum device or a vehicle inertia memory system that can detect the slow speed of the vehicle, but brake pressure-sensitive devices, and particularly hydraulic-electric switches have important advantages in the system, particularly since they can respond whenever the skid condition has passed, regardless of the speed of the car, to enable normal operation, while offering low speed protection against the governor's cutting off the brakes completely. The adjustment of the pressure level to which the device responds can be adjusted according to the wishes of the operator and the type of road conditions to be expected. But, in fact, a single setting, below that at which skidding on ice occurs, which is determinable for any given vehicle, is advantageously employed to give anti-skid protection under virtually all conditions.

Any tendency for the unit 24 to erroneously sense a skid condition and actuate switch 234 in response to a rapid deceleration is overcome by use of properly calibrated weights 208 and a properly tensioned spring 230. In addition, I have found that an erroneous sensing is most likely to occur where there is a rapid acceleration of the drive shaft quickly followed by a rapid deceleration. The frictional brake or drag that is resiliently exerted on spool 220 by plunger 190 helps to overcome this problem since this action tends to reduce the torque applied to shafts 200 and 202 by the governor assembly. It has been found, however, that at high speeds the frictional engagement between plunger 190 and spool 220 results in undue wear of the parts. Therefore, plunger 190 is assisted by spring 230, and the plunger is disengaged at high speeds. To this end, stop means 198 engages flange 196 so as to limit the outward movement of plunger 190 whereupon no engagement exists between plunger 190 and spool 220 at relatively high speeds. I have realized that the crucial problem of rapid acceleration followed by rapid deceleration occurs at low speeds in the low gear range with conventional automobiles, and I have, therefore, found that it is not necessary to have plunger 190 exert a frictional drag on spool 220 and shaft 200 at relatively high speeds.

Although the operation of my invention is thought to be clear from the foregoing description, a brief resume of the operation will now be given. Under normal operating conditions, solenoid 188 is de-energized whereby the spring-loaded solenoid plunger maintains the skid sensing and control unit inoperative. The de-energized solenoid further insures that rotary valve 126 is in the position wherein atmosphere is introduced to chamber 96. Since

chamber 96 is also open to atmosphere, it follows that there is a pressure balance on each side of large piston 84 thus enabling spring 100 to take over and move the piston 84 to the position illustrated in FIG. 2. In this position, piston 68, which is connected to piston 84 and moves therewith, engages ball valve 64 so as to unseat same against the action of spring 62. With ball valve 64 so opened, fluid pumped from master cylinder 14 by brake pedal 12 is free to pass directly to hydraulic lines 78 and 80 to actuate the vehicle brakes. The pressure-sensitive switch is adapted to be closed by a low level of brake pressure, e.g. 120-125 p.s.i., placing the anti-skid unit into readiness for operation.

Upon encountering a skid condition, i.e., a condition wherein the vehicle operator applies more brake pressure than is required for a particular road surface, the vehicle wheels slow suddenly. This causes corresponding deceleration of drive shaft 142, and, through suitable gearing, drive gear 170 and cam gear 222 also slow abruptly. The inertia of the governor assembly causes the gear 222 to move longitudinally by the cam surface aforedescribed, said longitudinal movement of gear 222 causing switch 234 to be closed which results in energization of solenoid 188 and the retraction of solenoid plunger 190. The second switch or terminal 266 is engaged upon retraction of the solenoid plunger to maintain the solenoid energized until switch 18 opens, even though switch 234 has now opened. Switch 18 is preset, e.g. at 85–90 p.s.i. to be opened at a brake pressure below that at which skids can occur, e.g. 100–150 p.s.i. for ice, whereupon solenoid 188 will remain energized until possibility of skidding no longer exists, and the solenoid is then de-energized to restore normal foot brake action before the governor has a chance ever to totally relieve the brakes.

Regarding the operation of the governor, once the solenoid 188 has been energized, the governor control unit takes over. More specifically, when a skid condition is approached, the sudden slowing of the vehicle wheels causes the governor assembly to collapse under the action of spring 218, thus moving shaft 200 and spool 220 to rotate valve 126, this movement now being permitted due to retraction of plunger 190. Rotation of valve 126 blocks the flow of atmosphere to chamber 98 and simultaneously allows the introduction of a vacuum to said chamber. The introduction of vacuum overcomes spring 100, thus resulting in movement of pistons 84 and 68 to the left, viewing FIG. 2. This movement closes isolation valve 64, thus preventing the application of further fluid pressure by brake pedal 12. As the valve remains in this position, the degree of vacuum increases, causing further leftward movement of piston 68 (FIG. 3), so that pressure in lines 78 and 80 and the brakes are relieved. A point is reached when the brakes are relieved sufficiently to allow the wheels to speed up from their slowed condition, the degree of relief required depending on the degree of slipperiness of the road. Then shafts 200 and 202 speed up once again, whereby the governor assembly retracts shaft 200 and spool 220 to once again rotate valve 126 to the position where atmosphere is introduced to chamber 98 and the piston 68 moves to the right and increases the brake pressure, until a part is reached where the wheels slow again and the cycle repeats. This action continues in a hunting manner so long as the vehicle is in an incipient skid, thus insuring that maximum brake pressure is continually applied to the wheel brakes short of pressure that would cause locking of the wheels. Once the brake pedal is released or the brake pressure in the system has dropped below the minimum possible value to create a skid, hydraulic pressure switch 18 breaks the circuit to solenoid 188, whereupon the latter becomes de-energized and the brakes once again become under complete control of the operator.

It is important to note that although loss of vacuum in the vehicle system will render the anti-skid means ineffective, said loss of vacuum will in no way affect normal

operation of the vehicle brakes. Also, it is important to note that this system can be used with foot pedal controlled brakes of both the powered and nonpowered types.

While a spring-loaded fly ball type of governor has been illustrated, and is presently preferred, the broad concept of the invention include the use of other governors, all rotary weight governors, such as gravity loaded fly ball, centrifugal shaft and inertial shaft governors, as well as governors that operate through electrical action.

The concepts are likewise applicable to other brake systems including air brakes in which valve 126 could directly modulate air brake pressure. Likewise it is possible to separate the governor from the valve, let the governor actuate a switch and let the valve be actuated by a solenoid responsive to the condition of the switch.

While use with engine vacuum is very efficient and inexpensive, and unlike previous systems I have proposed, does not cause the engine to run roughly due to loss of vacuum, the invention is applicable to other sources of pressure differential, incompressible fluid as well as gaseous fluid and numerous aspects of the invention are useful with nonfluid types of brake systems.

Referring now to FIGS. 7 and 8, a slightly different form of the invention is disclosed in which the relieving device is incorporated as an integral part of a power brake unit. The system generally comprises a brake pedal 300, a master cylinder 302, an accumulator 304, a safety valve 306, a hydraulic pressure switch 308, a power brake control unit 310, and an anti-skid unit 312.

Actuation of brake pedal 300 operates piston 314 to pump fluid from cylinder 316 to conduit 318. Master cylinder 302 further comprises a fluid reservoir 320 communicating with cylinder 316 by means of passageway 322. A conduit 324 extends from reservoir 320, the function of which will hereinafter be described. Accumulator 304 is in communication with conduit 318 and comprises a spring loaded piston 326 and acts as a cushion and shock absorber for the fluid system where variations in fluid pressure occur.

During normal operation of the vehicle with which the power brake unit is associated, and with the brakes in their non-actuated position, the power brake unit 310 is in the position illustrated in FIG. 8. In this position, chambers 328 and 330 of cylinder 332 are both under vacuum, thereby equalizing the pressure on opposite sides of piston 334. This enables spring 336 to urge the piston 334 to the leftward position illustrated, said piston 334 being connected by means of rod 338 to a brake actuating piston 340. Piston 340 is slidably mounted in cylinder 342, said cylinder having an outlet conduit 344 leading to safety valve 306. Conduit 344 is also in communication with hydraulic pressure switch 308.

A supply of hydraulic fluid is located in cylinder 342 whereupon movement of piston 334 against the action of spring 336, and corresponding movement of piston 340 will force fluid from cylinder 342 under pressure through conduit 344 and with valve 306 in the position illustrated in FIG. 8 to conduit 346 and then to hydraulic lines 348 which lead to the wheel cylinders (not shown) of the vehicle's braking system. Thus it will be seen that movement of pistons 334 and 340 to the right, viewing FIG. 8, will cause operation of the vehicle brakes.

As hereinbefore indicated, under normal operating conditions of the vehicle, and with the brakes not applied, pistons 334 and 340 are in the position illustrated in FIG. 8 due to the fact that a vacuum exists in both chambers 328 and 330. More specifically, suction from the intake manifold 350 of the vehicle passes through conduit 352 into chamber 330 and then through passageway 354, through openings 356 in diaphragm 358 to conduit 360, through rotary valve 362 of the anti skid unit 312, through conduit 364 and then to chamber 328. The control unit generally shown at 366 which forms a part of power brake unit 310 is of conventional form and comprises the aforesaid diaphragm 358, hydraulically movable by means

of piston 368 to a closed position wherein the diaphragm engages the fixed seat 370 to block communication between passage 354 and conduit 360. The piston 368 also controls movement of poppet valve 372 normally maintained closed by spring 374. Spring 376 normally urges diaphragm 358 to its open or unseated position, as illustrated in FIG. 8. An atmosphere intake is located at 378, and introduction of atmosphere into the system is controlled by valve 372 in a manner now to be described.

As above indicated, with the parts in the positions illustrated in FIG. 8, the brakes of the vehicle are non-operative due to the fact that suction is free to circulate through conduit 352 to chamber 330 and then through control unit 366 to conduit 360, through rotary valve 362 to conduit 364 and then to chamber 328. With vacuum on both sides of piston 334, spring 336 maintains the piston 334 and the contacted brake actuating piston 340 in brake release position. Upon actuation of brake pedal 300 by the operator, pressurized fluid passes through conduit 318 to conduit 380. With the safety valve 306 in the position illustrated, flow of the fluid into the valve is blocked, and hence the brake fluid is forced to flow through conduit 380 into the brake control unit 366. None of the fluid from the pedal is allowed to reach the brake actuators at 348.

Introduction of the pressurised fluid to the brake control unit 366 causes actuation of piston 368 to move diaphragm 358 to its closed or seated position and to simultaneously open poppet valve 372 in a conventional manner. This movement blocks communication between chamber 330 and conduit 360 and simultaneously allows air to enter through inlet 378 through the now open valve 372 to conduit 360. The atmosphere circulates around conduit 360 through valve 362 to conduit 364 and then the chamber 328. The introduction of atmospheric pressure to chamber 328 creates a pressure differential which overcomes spring 336 and moves piston 334 and piston 340 to brake actuating position. More specifically, the movement of piston 340 in cylinder 342 forces the brake fluid through conduit 344 and safety valve 306 to conduit 346 and hydraulic brake lines 348 to actuate the vehicle brakes in known fashion.

In order to insure that there is always an adequate supply of brake fluid in cylinder 342, conduit 324 connects fluid reservoir 320 to the cylinder 342, it being important to note that the inlet 382 to the cylinder 342 is located behind the forward portion of piston 340. Piston 340 is provided with a one-way valve 384 which enables fluid to pass through the piston from left to right, viewing FIG. 8, but which blocks flow of fluid through the piston from right to left. Thus, during operation of the piston 340, the suction created thereby in cylinder 342 sucks fluid through conduit 324 from reservoir 320 into cylinder 342 on the rear side of piston 340. During the return stroke of piston 340, the valve 384 allows fluid to pass through the piston, thereby insuring that there is always a full supply of fluid in the cylinder for brake actuation.

Since the structure and operation of hydraulic pressure switch 308 and anti-skid unit 312 are identical to that already described in connection with the first form of my invention, it is not thought that further description is necessary. It will be noted, however, that where the operator applies more brake pressure than is required for a particular road surface, the vehicle wheels will lock, whereby the same structure and operational sequence will cause rotary valve 362 to be moved to a position wherein conduit 360 becomes blocked and conduit 352A becomes open to allow suction from manifold 350 to be introduced through conduit 364 to chamber 328, whereupon spring 336 will once again take over to release the brakes. The same modulating section as aforedescribed will take place until hydraulic pressure switch 308 eventually renders the anti-skid inoperative.

In accordance with my invention it is crucial that the entire brake pressure be controlled by the anti-skid unit

and that no fluid be conducted directly from the foot pedal to the wheels. Means are provided to insure that brake pressure will not be lost should there be a loss of suction. This is accomplished by safety valve 306. More specifically, conduit 352B is connected to the suction conduit 352 whereupon as long as proper suction exists in the system, piston 386 is maintained in the position illustrated in FIG. 8 against the action of spring 388. In this position, conduit 344 is in communication with conduit 346, but conduit 350 is blocked at its extremity adjacent the safety valve. Should suction be lost in the system, spring 388 will become effective to move piston 386 upwardly, thereby blocking the communication between conduits 344 and 346 and at the same time providing communication between conduit 350 and conduit 346. With the safety valve in this position, the brakes are directly controlled by operation of pedal 300 since the entire power unit is now bypassed.

A one-way check valve 390 is located in conduit 352 adjacent the intake manifold 350 since without some backfire of the vehicle engine would cause an explosive pressure to be introduced to the power unit, which would be undesirable for obvious reasons.

Referring once again to FIG. 4, it will be noted that should the vehicle move in reverse, the direction of rotation of drive shaft 142 will likewise be in reverse, thus causing reverse rotation of flexible shaft 164 and shaft 168. Reverse rotation of shaft 168 will, in turn, cause reverse rotation of drive gear 170 and cam gear 222, causing pin 228 to leave stop surface 227 and move toward stop surface 225, which, as previously described, causes gear 222 to slide on shaft 202 in a direction away from partition 180 to close switch 234 and thus activate the governor control and anti-skid means. Even though it is possible that very slow reverse movement of the vehicle might not be sufficient to cause movement of cam gear 222 to activate the governor control, the fact remains that there is always the possibility and even likelihood that reverse movement of the vehicle will activate the governor control even though no skid condition exists. Since there is no practical need for any anti-skid means when the vehicle is moving in reverse, and since the activation of these means when not needed might even result in the loss of some stopping distance, it has been found desirable to provide means for rendering the governor control and anti-skid ineffective and inoperative when the vehicle moves in reverse.

Referring to FIGS. 9 through 12, a somewhat modified anti-skid unit is shown generally at 400. Unit 400 is basically the same as skid-sensing and control means 24 hereinbefore described except that a reverse cut-off switch has been incorporated in the unit to prevent energization of solenoid 188 when the vehicle drive shaft is in reverse and flexible shaft 164 correspondingly rotates in a reverse direction. The reverse cut-off switch, shown generally at 402 (FIG. 10) comprises a shaft 404 slidably coupled to shaft 164 as at 406, in the same manner that shaft 168 was drivingly engaged to flexible shaft 164. Shaft 404, which is constructed of any electrically conductive metal, is journalled in bore 408, said bore having an enlarged end portion 410, it being understood that the larger diameter of portion 410 results in the presence of an annular shoulder 412 at the extremity of bore 404. Slidably mounted in portion 410 is a contact button 414, said button being resiliently urged toward bore 404 by means of spring 416. The opposite end of spring 416 resiliently engages a contact cup 418 having a terminal 420 extending radially therefrom.

Adjacent its opposite extremity, shaft 404 has fixedly mounted thereon the aforedescribed drive gear 170, said gear meshing with cam gear 222, all in the manner hereinbefore described. A spring 422 engages washer 424 to normally urge shaft 404 to the left when viewing FIGS. 9 and 10, thus causing the end of shaft 404 to extend into enlarged portion 410 and, therefore, into engagement with contact button 414, as illustrated in FIG. 9. Loosely

mounted on shaft 404 is a reverse switch cam 426 comprising a circular plate 428 having a spiral cam portion 430 mounted thereon, and further having an outwardly extending pin 432 located adjacent the end of cam track 430. A cam pin 434 extends radially from shaft 404 and is adapted to cooperate with the spiral cam track 430 to determine the axial position of shaft 404. Expressed differently, when shaft 404 is rotating in a counterclockwise direciton when viewing FIG. 12, cam pin 434 will engage the stop pin 432 at the low point on the spiral track 430. With the parts in this position, shaft 404 will assume the position illustrated in FIG. 9 wherein the extremity of the shaft is in engagement with contact button 412, it being understood that spring 422 is somewhat stronger than spring 416, whereupon the extremity of shaft 404 will actually enter into enlarged portion 410. Upon clockwise rotation of shaft 404, pin 434 will ride up the spiral cam track 430 until it reaches the high end of said track and engages the opposite side of stop pin 432, as illustrated in FIG. 10. In this position, shaft 404 has been moved to the right (when viewing FIG. 10) against the action of spring 422 sufficiently for the end of said shaft to become completely withdrawn from enlarged portion 410. Since the annular shoulder 412 prevents the contact button 414 from moving out of enlarged portion 410, it follows that the end of shaft 404 becomes disengaged from contact button 414, as clearly illustrated in FIG. 10. Also mounted on shaft 404 is a washer 436 having a radially extending contact arm 438, the purpose of which will hereinafter be described. As will be seen most clearly in FIG. 10, the washer 436 is mounted on shaft 404 adjacent the flat rear surface of cam 426, it being understood that shaft 404 may freely rotate within washer 436. A cam thrust washer 440 comprising a peripheral ring portion 442 and bent marginal lugs 444 resiliently engages the opposite face 428 of cam 426, whereupon to resiliently urge cam 426 into engagement with washer 436 and at the same time to impart a sufficient frictional drag on cam 426 so that movement of shaft 404 will not impart corresponding rotary movement to the cam until pin 434 moves into engagement with stop pin 432. The thrust washer 440 has a radially extending spring arm 446 that is secured to the housing 176 by any suitable means, such as by having the end of the spring arm clamped between adjacent portions of the housing, as illustrated in FIGS. 9 and 10.

The operation of reverse switch 402 is as follows. Assuming that shaft 404 is being rotated in a counterclockwise direction, which for purposes of this description corresponds to forward movement of the vehicle, the switch will, as afore stated, assume the position illustrated in FIG. 9 wherein shaft 404 is making engagement with contact button 412. With the parts in this position, and assuming that a skid condition commences to exist, causing cam gear 222 to close switch 234, current will flow through line 262 to terminal 420 through contact cup 418, spring 416, contact button 414, and then to shaft 404, which as previously described, is in engagement with contact button 414. Shaft 404 is in turn in electrical engagement with washer 436 and its radially extending terminal 438, both directly and through the engagement of pin 434, which in turn engages cam 426, which in turn bears against washer 436. Thus, the flow of current is through terminal arm 438 and then to line 262A, which connects to ground as at 264. This, along with line 256, completes the circuit to solenoid 188 in order to energize same and render the governor control effective, all in a manner hereinbefore described. On the other hand, should the vehicle move into reverse, thus resulting in clockwise rotation of shaft 404, cam 426 will cause the shaft 404 to retract and become disengaged from contact button 414, as illustrated in FIG. 10, thus interrupting the connection to ground and preventing energization of solenoid 188. It will thus be seen that whenever the vehicle and its drive shaft moves in reverse, the reverse cut-off switch 402 will automatically prevent the skid sensing and control means and the gov-

[PX 12, p 16]

ernor control from coming into play. Thus, the vehicle brakes will always be under normal and direct control by the operator whenever the vehicle is moving in reverse.

While there is shown and described herein certain specific structure embodying the invention, it will be manifest to those skilled in the art that various modifications and rearrangements of the parts may be made without departing from the spirit and scope of the underlying inventive concept and that the same is not limited to the particular forms herein shown and described except insofar as indicated by the scope of the appended claims.

I claim:

1. An anti-skid device comprising a valve for controlling the relief and application of brake pressure, and control means responsive to skid conditions for actuating said valve, said control means capable of producing only a limited actuating movement and said valve comprising a rotary valve member adapted to alternately connect a brake control conduit to first and second passages maintained at different fluid pressures, said valve member comprising an elongated rod, first and second axially spaced-apart passageways through said rod, each of said passageways having a cross-section elongated in the axial direction of a length exceeding the diameter of said rod, said passageways being angularly displaced from one another about the circumference of the rod by less than 45°, said rod in one position adapted to connect said conduit with said first passage while sealing said second passage, and in a second rotary position adapted to connect said conduit with said second passage while sealing said first passage and means connecting said valve member for rotary movement between said positions in response to said actuating movement of said control means.

2. Valve means for use with a vehicle wheel anti-skid system in a vehicle brake system having a brake fluid source, a vehicle wheel brake operating actuator cylinder in fluid communication with said brake fluid source, operator controlled means for applying pressure to said vehicle wheel brake operating actuator cylinder, said vehicle wheel anti-skid system being interposed between said vehicle wheel brake operating actuator cylinder and said operator controlled means and including means for sensing a predetermined deceleration of the vehicle wheel, means for controlling the valve means, means for alternately applying and relieving the brake fluid pressure applied to the vehicle wheel brake operating actuator cylinder, said means for alternately applying and relieving the brake fluid pressure applied to the vehicle wheel brake operating actuator cylinder including a piston cylinder, a pressure operated piston mounted in said piston cylinder and means for connecting one side of said piston alternately to a first fluid pressure source or a second fluid pressure source said last named means including said valve means, said valve means comprising a first conduit fluidly connected to the first fluid pressure source and a second conduit fluidly connected to the second fluid pressure source and a third conduit fluidly connected to the piston cylinder, a rotatable member, said member being formed with a first bore and a second bore, said member being mounted fluidly to connect the first fluid pressure source to said third conduit through said first bore and first conduit in a first member position and fluidly to connect the second fluid pressure source to said third conduit through said second bore and second conduit in a second member position, and said bores being spaced from one another about the circumference of the rotatable member a distance sufficiently minimal to provide, in intermediate positions between said first and second member positions, for simultaneous fluid connection of both of said fluid sources with said third conduit.

3. Valve means as set forth in claim 2 wherein said member is a rod and said bores are axially spaced along said rod and said first conduit is aligned with and in fluid communication with said first bore and said second bore is cut off from fluid communication with said second conduit in the first member position and said second conduit is aligned with and in fluid communication with said second bore and said first bore is cut off from fluid communication with said first conduit in said second member position.

4. Valve means as set forth in claim 3 including a pin secured to one end of said rod, the means for controlling said valve means being operatively connected to said pin whereby upon actuation of said control means said pin is moved thereby to rotate said rod from one to the other of said positions.

5. Valve means as set forth in claim 4 including a housing, said rod being rotatably mounted in said housing, said housing having formed therein first, second and third chambers, said first conduit being fluidly connected to said first chamber, said first bore being fluidly connected to said first conduit in said first member position, said second conduit being fluidly connected to said second chamber, said second bore being fluidly connected to said second chamber in said second member position, said third conduit being fluidly connected to said third chamber, said first bore being fluidly connected to said third chamber in said first member position and said second bore being fluidly connected to said third chamber in said second member position.

6. Valve means as set forth in claim 5 wherein the second fluid pressure is higher than the first fluid pressure and the means for controlling said valve means holds said rod in the second member position until the sensing means has sensed the predetermined deceleration of the vehicle wheel after which said rod is rotated by said controlling means alternately between said first and second member positions.

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CLARENCE R. GORDON, Primary Examiner

U.S. Cl. X.R.

137—53, 54, 625; 303—21

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[PX 12, p 3]

[PX 13, p 1]

PLAINTIFF'S EXHIBIT 13

Nov. 11, 1969 F. A. PERRINO 3,477,765

ACCELERATION RESPONSIVE DEVICES FOR ANTI-SKID UNITS

Original Filed Nov. 2, 1964 7 Sheets-Sheet 1

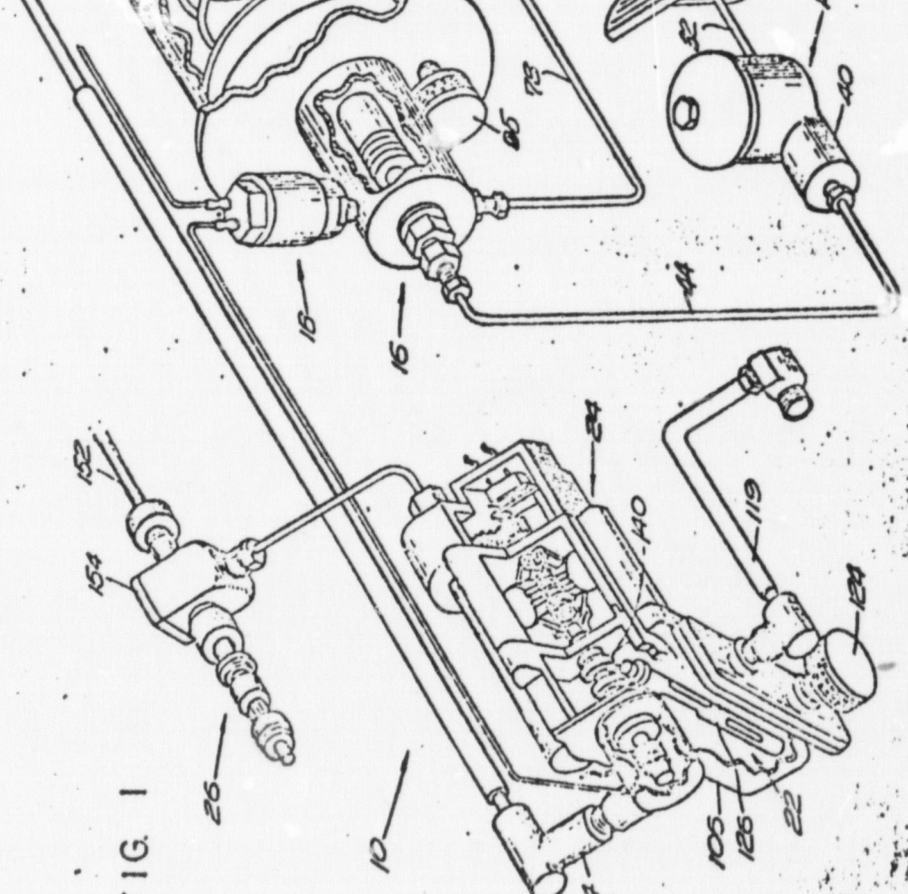


FIG. 1

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[PX 13, p 2]

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ACCELERATION RESPONSIVE DEVICES FOR ANTI-SKID UNITS

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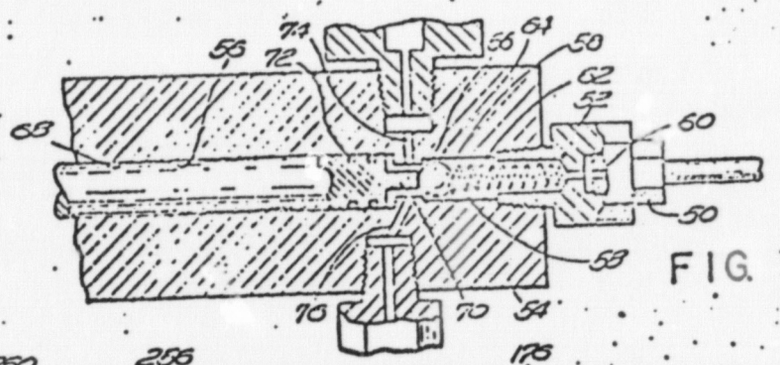


FIG. 3

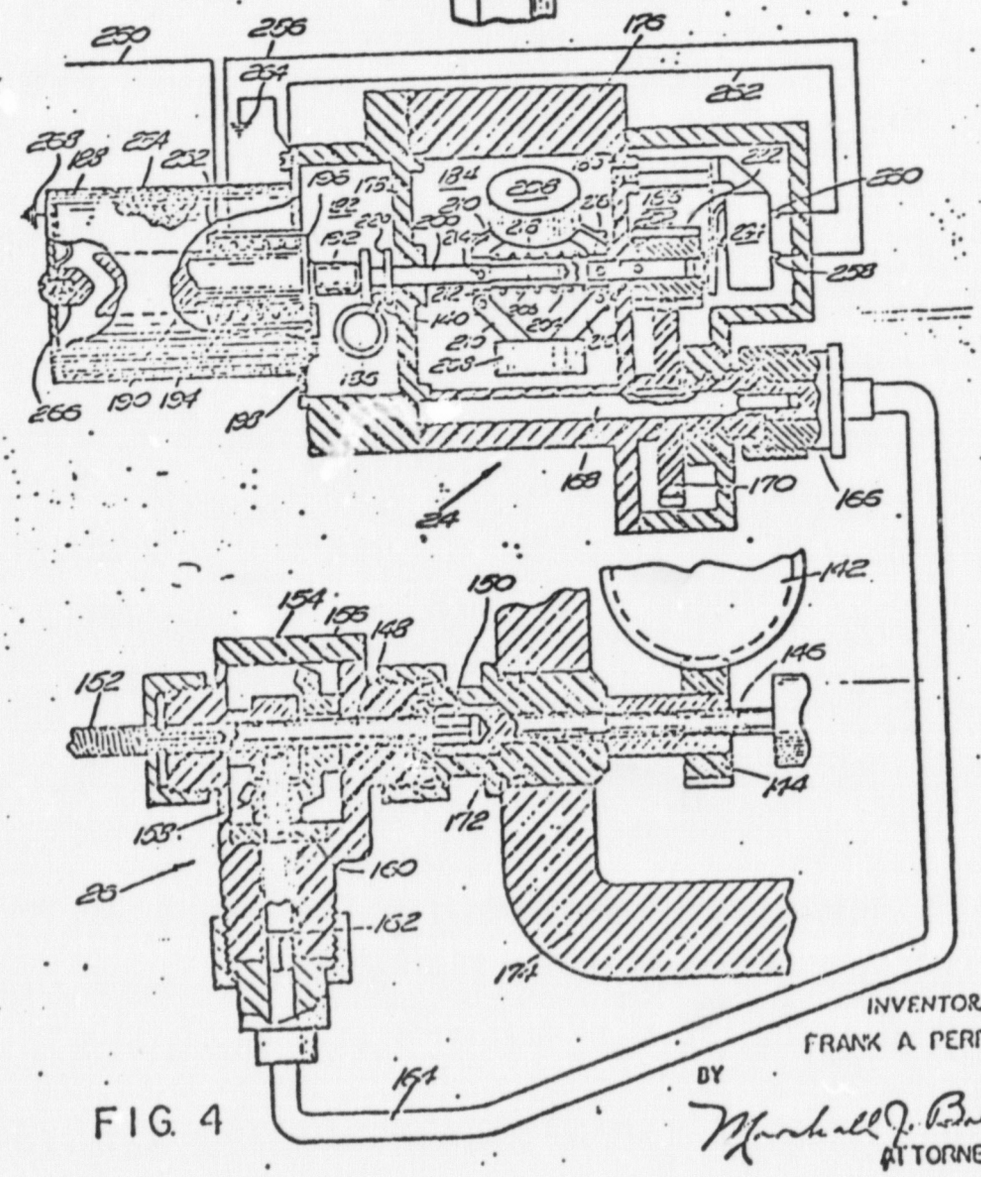


FIG. 4

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FIG. 2

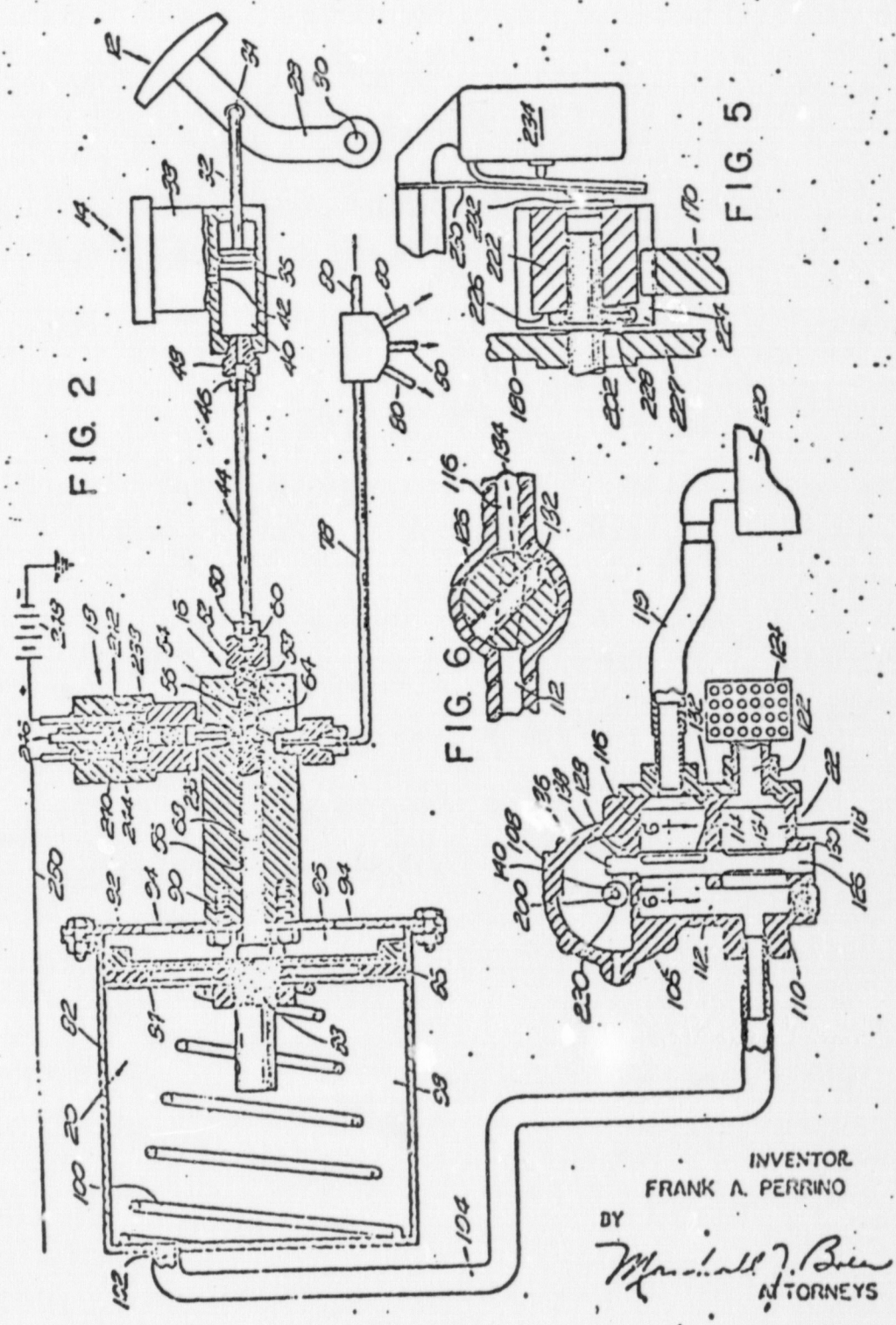


FIG. 5

FIG. 6

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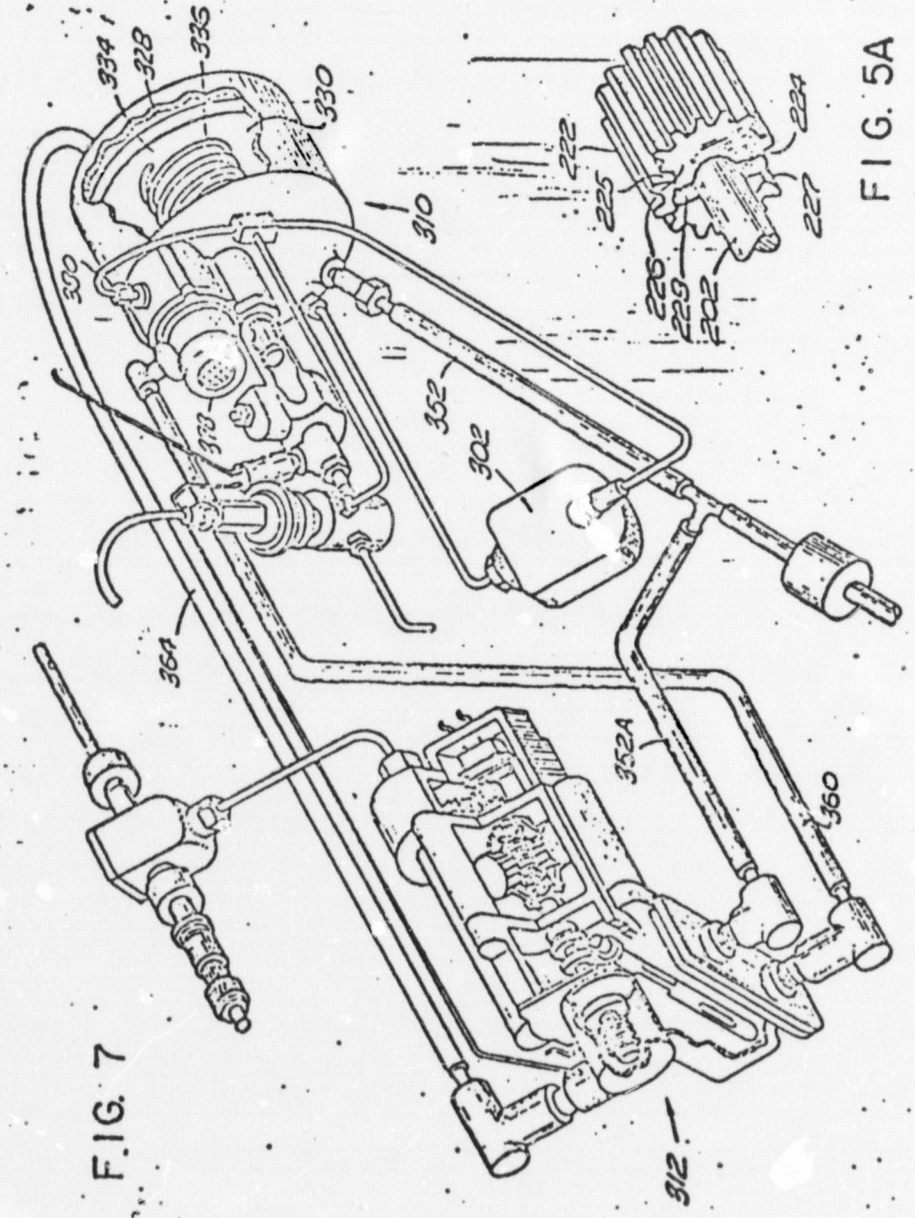


FIG. 5A

FIG. 7

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FIG. 8

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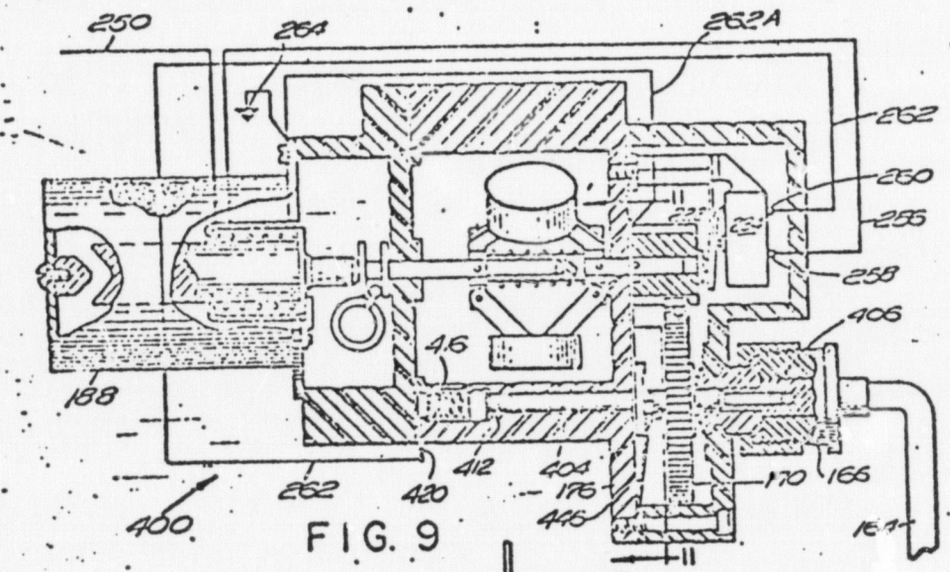


FIG. 9

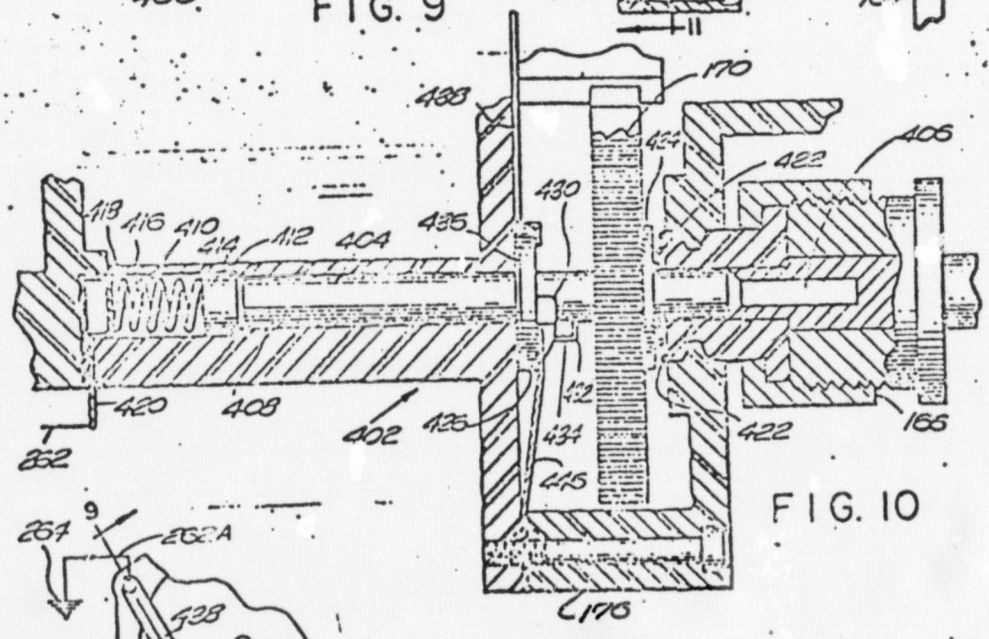


FIG. 10

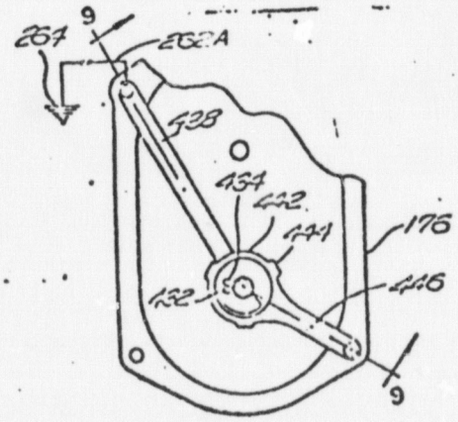


FIG. 11

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[PX 13, p 7]

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ACCELERATION RESPONSIVE DEVICES FOR ANTI-SKID UNITS

Original Filed Nov. 2, 1964 7 Sheets-Sheet 7

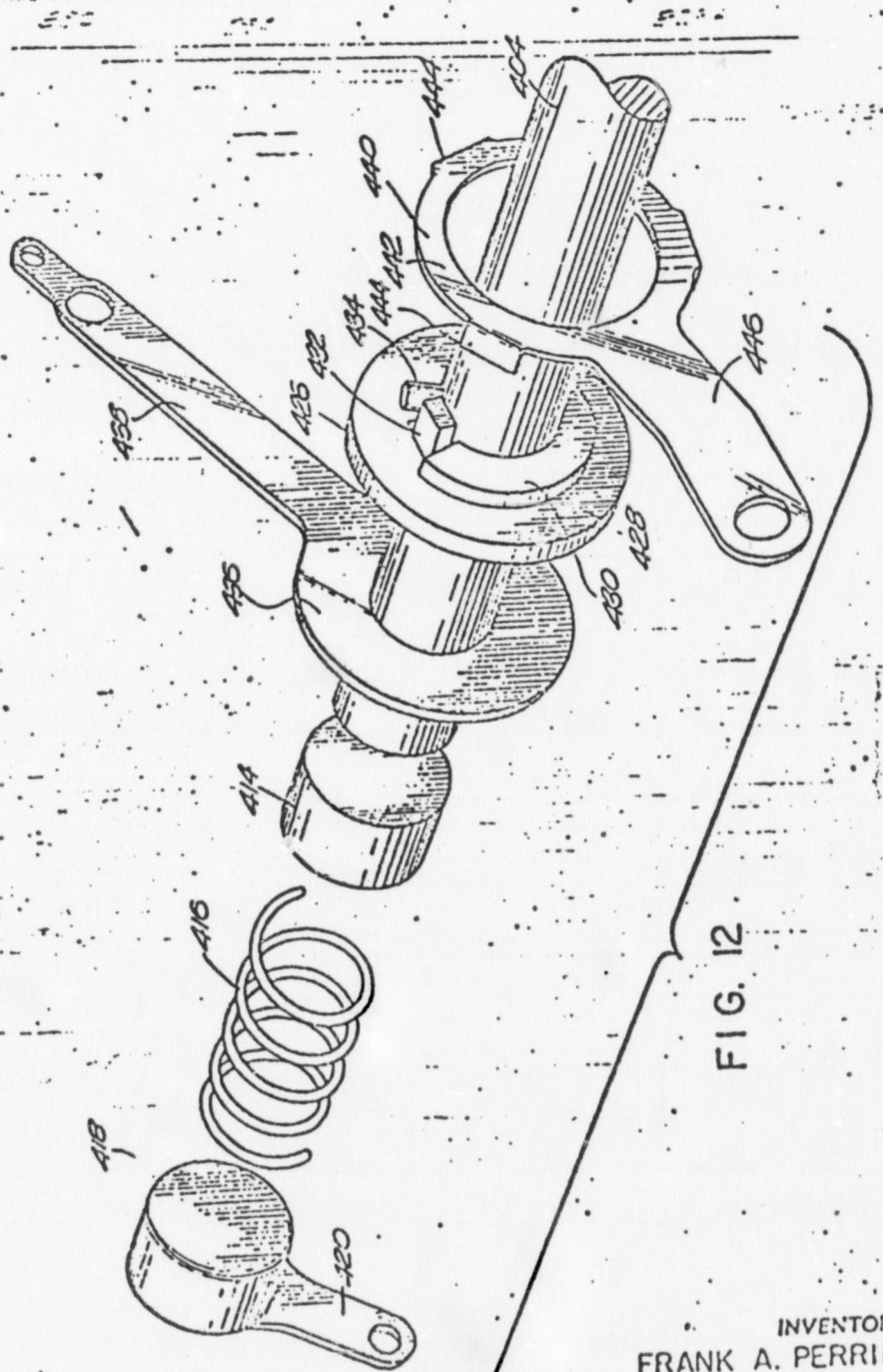


FIG. 12

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United States Patent Office Patented Nov. 11, 1969

3,477,765

ACCELERATION RESPONSIVE DEVICES FOR ANTI-SKID UNITS

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Original application Nov. 2, 1964, Ser. No. 409,350, now Patent No. 3,325,226, dated June 13, 1967. Divided and this application Aug. 5, 1966, Ser. No. 581,676

Int. Cl. B60t 8/24

U.S. Cl. 303—21 4 Claims

ABSTRACT OF THE DISCLOSURE

An anti-skid brake system in which a sensing means for sensing when a skid condition begins is caused to activate a wheel speed-sensitive device that takes over control of the brake pressure until the skid condition ends. The sensing means is an inertia-responsive means that senses a sudden change in the speed of rotation of a wheel at the commencement of skid conditions. The control of the brake pressure is taken over completely by a mechanical governor driven dependently with the wheel. The governor then controls the brake pressure in a hunting or modulating manner, for an extended period, until such control of the brakes is ended by a reset means that responds to slowing of the vehicle to a predetermined speed. Preferably, the reset means is a brake-pressure-sensitive device which senses reduction of brake pressure to a predetermined level which the governor can produce only when it slows to the predetermined speed.

This is a division of application Ser. No. 409,350, filed on Nov. 2, 1964, and entitled Braking System and Anti-skid Means Therefor, now U.S. Patent No. 3,325,226.

The present invention relates generally to braking systems for automotive vehicles and the like and is particularly concerned with the provision of novel and improved anti-skid means.

It is well known that a skid condition may be encountered by a moving vehicle where the brakes of the vehicle are applied so hard that the vehicle wheels tend to lock. Quite obviously, other factors affect the commencement and existence of a skid condition, such as the condition of the surface on which the vehicle is moving, the speed at which the vehicle is traveling, etc. However, it is possible for a vehicle to commence a skid even where the surface on which the vehicle is traveling is not wet or icy and even where the vehicle is traveling at moderate speeds, if the vehicle brakes are applied too hard. Since all control of a vehicle is lost while the vehicle is in a skid, thus increasing the danger of serious accident and personal injury, it is obviously highly desirable to prevent or reduce skidding while at the same time enabling the vehicle to come to a complete stop as quickly as possible.

It is therefore a primary object of the present invention to provide an improved braking system wherein operation of the brakes is under control of the vehicle operator until a skid condition commences to exist, at which time anti-skid means automatically take over complete control of the brakes and maintain said control until danger of skidding no longer exists, whereupon the brakes are returned to normal control by the operator.

Another object of my invention is the provision of efficient and sensitive control means for the various operations of the system, which have usefulness in other types of systems as well.

Another object of my invention is the provision of novel and improved anti-skid means equally adaptable to so called power brakes, as well as nonpower brakes.

A further object of my invention is the provision of anti-skid means of the character described wherein the vacuum system of the vehicle is utilized, but wherein the loss of vacuum in no way interferes with control of the vehicle brakes by the operator.

A further object of my invention is the provision of anti-skid means that is readily and easily adaptable to existing vehicles.

Another object is the provision of a novel and improved reverse cut-off switch that automatically renders the anti-skid means inoperative when the vehicle moves in reverse.

Still another object of my invention is the provision of an anti-skid system for automotive vehicles and the like having means which are durable and effective in operation and which are feasible and practical both from an economic and manufacturing standpoint.

Other objects, features and advantages of my invention will become apparent as the description thereof proceeds when considered in connection with the accompanying illustrative drawings.

In the drawings which illustrate the best mode presently contemplated for carrying out the present invention:

FIG. 1 is a diagrammatic view, in perspective, of the assembly utilized in my nonpower unit, with portions being broken away for purposes of illustration;

FIG. 2 is an elevational view, in section, of a portion of the overall assembly illustrated in FIG. 1;

FIG. 3 is an enlarged fragmentary sectional elevation of a portion of the brake-actuating means illustrated in FIG. 1;

FIG. 4 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid unit and the drive means therefor;

FIG. 5 is an enlarged sectional elevation of the cam gear and solenoid-actuating switch which form a part of the anti-skid unit;

FIG. 5A is an enlarged fragmentary perspective view of the cam gear;

FIG. 6 is an enlarged section taken on line 6—6 of FIG. 2;

FIG. 7 is a diagrammatic view, in perspective, of my anti-skid means in combination with a power brake unit, portions being broken away for purposes of illustration;

FIG. 8 is an elevational view, in section, of the power brake assembly illustrated in FIG. 7;

FIG. 9 is an enlarged fragmentary sectional elevation illustrating certain details of the anti-skid means and the reverse cut-off switch in combination therewith;

FIG. 10 is an enlarged fragmentary sectional elevation of the reverse cut-off switch per se;

FIG. 11 is a fragmentary section taken on line 11—11 of FIG. 9; and

FIG. 12 is an enlarged exploded perspective view of the reverse cut-off switch per se.

According to one aspect of my invention, a sensing means for sensing when a skid condition begins is caused to activate a wheel speed-sensitive device that takes over control of the brake pressure until the skid condition ends. Advantageously, the sensing means is an inertia-responsive means that senses a sudden change in the speed of rotation of a wheel at the commencement of skid conditions. This inertia-responsive device does not control the brake pressure itself by repeated action during the skid condition, as has been attempted by the prior art. Instead the control of the brake pressure is taken over completely by a governor driven dependently with the wheel, advantageously a mechanical governor. The governor then controls the brake pressure in a hunting or modulating manner, for an extended period, until such control of the brakes is ended by a reset means that responds to slowing of the vehicle to a predetermined

speed. Preferably, the reset means is a brake-pressure-sensitive device which senses reduction of brake pressure to a predetermined level which the governor can produce only when it slows to the predetermined speed.

This separation of the initiating and controlling functions is an extremely important feature. The initiating device must necessarily have a restraint to prevent operation when the vehicle merely decelerates normally. I have realized that such a restraint prevents the device from being able to effectively regulate the brake pressure due to lack of sensitivity. The governor, after it takes over control, is free of such restraint and therefore can have much greater sensitivity with the result that the brake pressure can be much more accurately regulated.

The reset device is also of critical importance because at low speeds, e.g. below 10 m.p.h., the governor forces can become so diminished that the governor can behave entirely as if the vehicle were skidding, though not, thereby totally relieving the brake pressure. The reset device can act before this occurs, whereby normal control is regained.

A further aspect of my invention concerns the realization that a predetermined level of brake pressure, e.g. the minimum brake pressure at which skidding of the vehicle will occur on ice, generally between 100–150 p.s.i. for conventional automobiles, can be utilized to determine the setting for the brake pressure-sensitive reset device. Thereby, only at the time when skidding is no longer possible will control be taken away from the anti-skid unit to return the brake system to normal operation. This pressure-sensitive device, particularly when taking the form of a brake fluid pressure-sensitive switch, is useful with anti-skid systems generally, though of particular usefulness in the system described above.

Advantageously, the brake pressure-sensitive switch operates an electrical system, preferably a solenoid, to restore normal brake operation, and it can be employed to bring the anti-skid unit into readiness whenever the operator applies his brakes, as well as to stop the anti-skid unit from acting when skid conditions have ended.

Another aspect of the invention concerns the control of the governor. In normal operation of the brakes, the governor is prevented from influencing the brakes, but once activated, e.g. by the inertia-sensitive device, it is made continually to act, even when the vehicle wheel speeds up after the brakes have been initially relieved. A controlling or restraining member, preferably a solenoid, is employed to achieve this operation, with the controlling member held steadily from interfering with the governor until the reset device operates. In the case where a rotary weight mechanical governor is employed, a plunger can normally restrain the governor and a solenoid, initially under the control of the inertia-responsive device, can pull back the plunger to free the governor, whereupon the plunger can operate a switch to maintain the solenoid energized, and at the end of skid conditions the reset device can de-energize the solenoid to once again restrain the governor.

Another aspect of the invention is a fluid valving system that responds to the governor to increase and decrease brake pressure. Advantageously, the valve, discharging to a brake control conduit, is moved by the governor alternately between two positions to connect two passages for fluid at different pressures to the conduit. This causes the brake pressure to rapidly vary in a modulating or hunting manner, to keep the brake pressure at maximum values while preventing skidding. The mean about which the pressure modulates changes as the slipperiness of the road changes, due to the effect such road changes have upon the governer.

For use in a closed-circuit hydraulic system, the direct control of the brakes is achieved, according to the invention, by an isolation valve that continually seals the wheel brake from the control of the operator, and a variable volume chamber, e.g. piston and cylinder, for relieving and reapplying brake pressure to the wheel, the piston modulating about a mean determined by the modulating pressure conditions of the brake control conduit. Advantageously, the piston, at the position where it allows full pressure, can operate the isolation valve. Then the piston being spring loaded to that position can also perform a pressure-limiting function, limiting brake pressure to e.g. 1,000 p.s.i., though the operator is capable of applying twice that much. In normal operation the brakes function as well at 1,000 p.s.i. as above, and by so limiting the brake pressure, less pressure is present to be relieved when a skid occurs, which enables faster action and allows the relief chamber to be smaller.

In other brake systems the brake control conduit can control a power unit, or in the case of air brakes, it can directly control the brake actuators associated with the wheels.

Furthermore, according to the invention, I have provided advantageous and improved constructions of the various operating components and sub-systems which can best be described with reference to the drawings.

Referring now to the drawings, and more particularly to FIGS. 1–6 thereof, a closed hydraulic circuit braking system employing my novel and improved anti-skid means in combination with a so-called nonpower brake will be described. The over-all system, generally designated as 10, comprises brake pedal 12, master cylinder 14, brake-relieving device 16, hydraulic pressure switch 18, relief actuator means 20, conduit valve means 22, skid-sensing and control means 24, and anti-skid unit drive means 26.

Referring to FIGS. 1 and 2, the brake pedal 12 is of conventional construction and comprises bell crank 28 pivotally connected to a suitable support on the vehicle (not shown) at 30 whereupon depression of pedal 12 causes bell crank 28 to rotate in a counterclockwise direction around pivot point 30, thus actuating connecting rod 32 pivotally connected to bell crank 28, as at 34. Rod 32 carries at its other extremity a piston 36 which functions to pump oil or other suitable fluid from master cylinder 14, all in a well-known manner. Specifically, master cylinder 14 has an upper reservoir 38 which communicates with lower chamber 40 by means of opening 42 whereupon a full supply of oil is always present in the chamber 40 to be pumped by actuation of pedal 12 and piston 36.

Fluid pumped from master cylinder 14 passes through a conduit 44 connected at one of its extremities by suitable couplings 46, 48 to chamber 40 of master cylinder 14 and at its other extremity by a suitable coupling 50 to the brake-relieving device generally designated at 16, which incorporates an isolation valve. More specifically, coupling 50 is connected to a second coupling 52 which is threadedly attached to block 54, said block having a bore 56 extending longitudinally therethrough into communication with aligned longitudinally extending bores 58 and 60 in couplings 52 and 50, respectively.

As will be seen most clearly in FIGS. 2 and 3, coupling 52 is provided with an internal shoulder for receiving positioning spring 62 which positions isolation valve ball 64 adjacent valve seat 66, tending to close with slight force the bore or passageway 56. During normal operation of the vehicle, ball 64 is maintained in its unseated position, as illustrated in FIGS. 2 and 3, against the action of spring 62, by means of spring-loaded actuator piston 68 having a reduced extension 70 in engagement with ball 64. Suitable O-rings or the like 72 are provided to insure a tight sliding fit between piston 68 and bore 56 whereby said piston effectively blocks further flow of fluid through bore 56.

Block 54 is provided with a pair of oppositely disposed passageways 74 and 76, it being noted that said passageways communicate with bore 56 just behind valve seat 66, whereupon when ball 64 is in its closed position in engagement with valve seat 66, fluid in conduit 44 from the master cylinder will be blocked before reaching the

passageway 74 and 76. Passageway 74 communicates with hydraulic pressure reset switch 18 for reasons hereinafter to be described, while passageway 76 communicates by means of suitable couplings with conduit 78, which in turn communicates with the hydraulic brake lines 80 leading to the wheel hydraulic actuator cylinders (not shown) of the vehicle to actuate the brakes in well-known fashion. Thus it will be seen that with the parts in the positions illustrated in FIGS. 2 and 3, and particularly with piston 68 in the position illustrated, brake pedal 12 controls actuation of the vehicle brakes through a direct line. This is the normal operating condition of the vehicle.

The actuator for the relieving device 16, generally designated at 20 (FIGS. 1, 2) controls the movement of piston 68 and hence ball 64. This actuator comprises a housing or cylinder 82 having a large piston 84 slidably mounted therein, said piston having suitable seals 86 at its outer periphery to insure that the piston 84 will make a tight sliding fit within cylinder 82 whereby no leakage will exit around the piston. Piston 84 is threadedly connected, as at 88, to the aforementioned small hydraulic piston 68, whereupon movement of large piston 84 causes corresponding movement of piston 68. Housing 82 is secured to block 54 by any suitable means, such as screws 90, and the adjacent end wall 92 of the housing 82 is in free communication with atmosphere, as by ports 94. Suitable air filters, such as shown at 95 in FIG. 1, may be associated with the parts 94. It will thus be seen that the chamber 96 located on one side of large piston 84 is always at atmospheric pressure, while chamber 98, located on the opposite side of piston 84, may either be at atmospheric pressure or it may be vacuum charged, as will hereinafter be described in more detail. When the chambers 96 and 98 are both at atmospheric pressure, spring 100 resiliently urges pistons 84 to the position illustrated in FIG. 2, it being seen that the piston 84 has moved to the right as far as it can go. In this position, the piston 68 has also been axially moved to the position illustrated, in which position extension 70 (FIG. 3) has engaged ball 64 to unseat it from valve seat 66. Thus, where chambers 96 and 98 are both at atmospheric pressure, the vehicle brakes are under the complete control of the brake pedal 12. However, when chamber 98 is vacuum charged, the pressure differential between chambers 96 and 98 is sufficient to overcome spring 100, whereupon piston 84 will move to the left, viewing FIG. 2, thus causing corresponding movement to the left of piston 68. This in turn enables ball 64 to seat against valve seat 66, by means of spring 62, thus blocking the bore 56 and preventing actuation of the brakes by means of brake pedal 12. As pistons 84 and 68 move further to the left due to the presence of a vacuum or even a partial vacuum in chamber 98, hydraulic pressure in lines 78 and 80 and in the spring-loaded brake actuators associated with the vehicle wheels will be relieved, due to the fact that the fluid is now free to back up into that portion of bore 56 that has been vacated by movement of piston 68 to the left. It will therefore be seen that by introducing a vacuum or partial vacuum to chamber 98, fluid pressure in lines 78 and 80 will automatically be relieved, thereby relieving the brakes. By the same token, as soon as atmosphere is again introduced to chamber 98 to equalize the pressure in chamber 98, the brakes can once again be fully actuated by the pedal 12. It will be apparent that the amount of vacuum introduced to chamber 98 will determine the amount of reduction in brake pressure. More specifically, the greater the vacuum that is introduced to chamber 98, the greater will be the movement of piston 84 to the left (viewing FIG. 2) and at the same time there will be greater movement of piston 68 to the left, thus vacating more of bore 56 and providing greater relief of the hydraulic brake pressure. Similarly, given a certain amount of relief due to a certain amount of vacuum, then decrease in the amount of vacuum will increase the hydraulic brake pressure due to movement of the pistons to the right. The ability of my system to provide varying degrees of relief and restoration of brake pressure while the brake pedal remains isolated from the system is an important and advantageous feature.

It will be noted that during normal operation of the system, as heretofore described, i.e., where atmospheric pressure exists in both chambers 96 and 98, the piston 68 will function as a pressure limiter in the system. More specifically, since the piston 68 is resiliently held in the position illustrated in FIG. 2 by means of spring 100, it will back off when excess pressure is applied by foot pedal 12. In practice it has been found that in a conventional automobile pressure above 1,000 p.s.i. or at least above 1,500 p.s.i., depending on the type of vehicle, is unneeded for normal brake operation and hence spring 62 and 100 are preferably calibrated so as to enable ball 64 to seat when pressure in excess of that needed is introduced, thus preventing introduction of excessive pressure to the brakes. It will be obvious that the springs may be calibrated so as to provide a limit at any selected pressure.

The means for introducing either atmospheric pressure or vacuum to chamber 98 and for modulating the degree of vacuum therein will now be described. Referring to FIG. 2, it will be seen that cylinder housing 82 is provided with an opening 102 in communication with chamber 98, which opening communicates with a brake control conduit 104 that in turn connects with the conduit valve, generally designated at 22. More specifically, there is provided a housing 106 having an upper portion 108 which houses the skid-sensing and control means 24, and a lower portion 110 which houses the valve 22. The lower housing 110 comprises an inner chamber 112 having a partition 114 defining a vacuum passage 116 and an atmosphere passage 118. More specifically, vacuum passage 116 is in communication, by means of passage 119, with the intake manifold 120 of the vehicle. Thus, suction is constantly being applied to the passage 116. The passage 118, on the other hand, is in communication with atmosphere through port 122, to which a suitable air filter 124 may be attached, if desired. A rotary valve 126 is mounted for rotation in housing 106, it being noted that valve 126 is an elongated rod that extends vertically through upper partition 128, the aforesaid partition 114, and is journaled in bottom wall 130, the partition and wall being integral parts of a unique molded housing of anti-friction plastic. The valve 126 is provided with a pair of passageways or bores 132 and 134 extending diametrically therethrough, said bores being vertically spaced from each other, and having elongated cross-sections in the axial direction, longer than the diameter of the rod. Bore 132 is associated with passage 116 and bore 134 with passage 118. The bores are angular, disposed with respect to each other, as shown most clearly in FIG. 6. Thus, with the rotary valve 126 in the position illustrated in FIGS. 2 and 6, bore 134 is positioned so as to allow free passage of atmosphere from passages 122 and 118, bore 134 and chamber 112 to brake control conduit 104, whereby atmosphere will flow into chamber 98. At the same time, due to the angular displacement of bore 132, passage 116 will be closed by valve 126, thus preventing suction from manifold 120 and passage 119 from coming into contact with chamber 112 and brake control conduit 104. It will be understood that the only access from passage 116 and 118 to chamber 112 is through valve 126 and more specifically, the bores 132, 134 thereof. Should valve 126 be rotated by means hereinafter to be described, it will be seen that communication between passage 118 and chamber 112 will eventually be blocked, thus interrupting the flow of atmosphere to brake control conduit 104, while at the same time passage 116 will be in communication with chamber 112, whereupon a suction will be applied through brake control conduit 104 to chamber 98.

It is important to note that use of a rotary valve such as valve 126 is extremely advantageous where passage of a vacuum and atmospheric pressure, or more generally two different pressures, is being controlled due to the fact that the pressure differential offers a minimum of resistance to the turning movement of the small diameter valve. This is contrasted to a reciprocal or poppet type valve wherein movement of the valve is much more resisted by pressure differential.

As hereinbefore explained and described, introduction of a vacuum or partial vacuum to chamber 93 automatically tends to release brake pressure on the vehicle wheels. It has been found in practice that an angular displacement between bores 132 and 134 of approximately 37.5° gives best results in the operation of my system, and it will be understood that rotary valve 126 may not always be positioned so as to completely open or close the bores 132 and 134. Thus, different degrees of movement of valve 126 allow different brake fluid pressures to be exerted on the vehicle wheels, thus providing the proper braking pressures for different road conditions. In instances where the pressure on the brake actuator is the same at the beginning of a skid, (lesser initial pressure requires less relieving), the amount of relief of the brake pressure depends on the amount of travel of piston 84 to the left in chamber 93 which in turn is dependent on the amount of vacuum introduced to the chamber. Thus, where the vehicle is braked and skids on a dry surface which requires a greater brake pressure to create a skid condition, lesser movement of piston 84 to the left, and hence less vacuum, will be required to relieve the brake pressure sufficiently to prevent the vehicle wheels from locking, this pressure being referred to as "release" pressure. Conversely, where the vehicle is braked on a slippery surface such as ice, a lesser brake pressure is required to create a skid condition. Hence the valve 126 must open suction passage 116 to the brake control conduit 104 for a longer initial period to provide greater vacuum to chamber 93 in order to move piston 84 sufficiently to the left so that sufficient relief of the brake pressure will be effected to prevent the wheels from locking.

In accordance with the invention after release pressure is attained, with corresponding movement of the piston then the governor causes the degree of vacuum in conduit 104 to modulate, which modulates the brake pressure through a narrow range of intermediate pressures above and below the release pressure, without ever restoring the normal, higher brake pressure until the skid condition ends.

To thus operate the valve, the upper extremity of valve 126 (FIG. 2) is provided with a reduced portion 136 that extends upwardly through partition 128, said portion 136 having a pin 138 extending therethrough to which is attached a circular projection 140, it being seen that as a longitudinal thrust is exerted against projection 140, the valve 126 will be caused to rotate.

The means for controlling movement of valve 126; namely, the skid-sensing and governor control means and the drive means therefor, will now be described. Referring to FIG. 4, skid-sensing and governor control means is designated generally at 24, and the drive means therefor is designated generally at 26. Referring first to the latter, the vehicle drive shaft is shown at 142, it being understood that the drive shaft is directly connected to the wheels of the vehicle (not shown) whereupon when the vehicle wheels lock, the drive shaft also locks. The drive shaft drivingly engages speedometer gear 144, which in turn is fixed to shaft 146, which in turn is drivingly connected to shaft 148 by means of a coupling shown at 150, said shaft 148 connecting with speedometer cable 152. Gear box 154 through which shaft 148 passes and in which the shaft 148 is suitably journaled, houses drivingly engaged bevel gears 156, 158, it being understood that bevel gear 156 is keyed to shaft 148 and bevel gear 158 is keyed to shaft 160, whereupon rotation of shaft 148, transmitted thereto by the vehicle drive shaft 142, is transmitted to shaft 160. Shaft 160, in turn, is drivingly interconnected, as at 162, to flexible shaft 164, said flexible shaft being connected, as at 166, to the unit 24, and said flexible shaft being drivingly interconnected with a shaft 168 which has fixed thereto a drive gear 170. It will therefore be seen that the gear 170 is drivingly interconnected with the vehicle drive shaft 142, which in turn is connected to the vehicle wheels. Thus, rotation of the vehicle wheels will cause corresponding rotation of gear 170, and, conversely, sudden slowing of the vehicle wheels, such as would occur where the vehicle approaches a skid condition by the operator applying more brake pressure than is required for a particular road surface, will cause not only the main drive shaft 142 to suddenly slow, but also drive gear 170. Gear box 154, and adaptor 172, by means of which the gear box is connected to the engine housing 174, comprise an assembly which may be easily and readily adapted to a conventional vehicle for driving the sensing and governor control means now to be described.

The unit 24 (FIGS. 1, 4) comprises a molded plastic housing 176 having a pair of spaced partitions 178, 180 therein defining chambers 182, 184 and 186. A solenoid 188 is secured to housing 176 adjacent chamber 182 by any suitable means, the solenoid 188 having a plunger 190 with a reduced extension 192 resiliently urged into chamber 182 by means of spring 194. When solenoid 188 is not energized, spring 194 bears against flange 196 affixed to plunger 190 to resiliently urge extension 192 into chamber 182. Suitable stop means 198 mounted on the interior of chamber 182 engage flange 196 to limit resilient axial movement of plunger 190 and extension 192 at a predetermined point, for reasons hereinafter to be explained more fully. Journaled in the partitions 178 and 180 is a pair of aligned shafts 200 and 202, respectively. Shaft 202 is provided with an enlarged portion 204 having an internal bore for slidably receiving in telescoping relation a reduced extension 206 of shaft 200. Attached to shafts 200 and 202 is a governor assembly comprising a plurality of weights 208, each pivotally connected to a pair of links 210, which links in turn are pivotally connected to arms 212. The arms 212 located on one side of the weights 208 are affixed to shaft 200 as at 214, while the arms on the other side of the weights are affixed to shaft 202 as at 216. As hereinbefore pointed out, shaft 202 is journaled for rotation in partition 180 but is incapable of longitudinal movement. Shaft 200, on the other hand, is drivingly engaged with shaft 202 so as to rotate therewith, but said shaft 200 is axially slidable with respect to the said shaft 202. Since the governor assembly is affixed on one side to shaft 202 and on the other side to shaft 200, it will be seen that the speed of rotation of the shafts will determine their longitudinal position relative to each other. Spring 218 resiliently urges the governor assembly to its collapsed position wherein shafts 200 and 202 are axially extended with respect to each other, but as the rotational speed of shafts 200 and 202 increases, centrifugal force urges the weights 208 outwardly, thus moving shaft 200 toward the right, viewing FIG. 4, against the action of spring 218.

Fixed to shaft 200 at the extremity thereof located in chamber 182 is a spool member 220, which spool member is in engagement with projection 140 carried by the reduced upper extension 136 of rotary valve 126. Thus, axial movement of shaft 200 causes like movement of the spool 220 which in turn carries therewith projection 140 captured between the opposed walls of the spool to thereby impart rotary movement to valve 126.

The means for imparting rotational movement to shafts 200 and 202 and the governor assembly associated therewith comprises a cam gear 222 (FIGS. 4, 5 and 5A), which cam gear is part of the drive train, in driven engagement with gear 170. The cam gear 222 is mounted

brake unit 310 is of conventional form and comprises ... aforesaid diaphragm 358, hydraulically ... vable by means ... entire brake pressure be controlled by the ... unit

[PX 13, p 12]

on shaft 202 and is located in chamber 186. The cam gear is slidably mounted on shaft 202 and is in driving engagement therewith by means now to be described. A recess 224 having inclined or helical surface 226 is provided at the end of the cam gear adjacent partition 180, and a projection or pin 228, carried by shaft 202, extends into said recess in engagement with the cam surface 226. At either end of the recess is a stop surface, at the deeper end stop surface 227, adapted to engage pin 228 for accelerating the governor and at the shallower end stop surface 225 for engaging pin 228 to decelerate the governor. A leaf spring 230 mounted to the housing 176 by any suitable means engages a button 232 which is press fitted into the central bore of cam gear 222 to normally urge the latter axially toward partition 180, holding pin 228 against accelerating stop surface 227. However, when rotation of drive gear 170 and cam gear 222 is suddenly slowed, as by sudden slowing of the vehicle wheels upon the presence of a skid condition, the rotational inertia of shaft 202, especially due to the governor weights 208, causes shaft 202 to turn ahead of gear 222, causing pin 228 to leave stop surface 227 and move to stop surface 225. This slides gear 222 on shaft 202 in a direction away from partition 180 to activate governor control.

It is important to note that solenoid spring 194 is stronger than governor spring 218, whereupon when solenoid 188 is not energized, plunger extension 192 bears against spool 220 to restrain same in the position illustrated in FIG. 4 wherein the rotary valve 126 is in the position illustrated in FIG. 2, namely, the position where atmosphere is free to pass through the valve to conduit 104 and then to chamber 98 and wherein at the same time suction from the intake manifold is blocked from communication with conduit 104. This is the normal operating condition of the system wherein the balanced pressure on opposite sides of actuator piston 84 enables spring 100 to position piston 84 and piston 68 as illustrated in FIG. 2. In this position, ball valve 64 is unseated to allow fluid pressure to pass directly to the brake line whereupon operation of the brakes is under direct control of brake pedal 12. This condition will remain so long as solenoid 188 (FIG. 4) is not energized. Energization of solenoid 188 is accomplished in the following manner. As previously explained, sudden slowing of gear 170 due to sudden slowing of the vehicle wheels as a result of a skid condition causes cam gear 222 (FIGS. 4, 5, 5A) to move axially on shaft 202 away from partition 180 against the action of spring 230. This movement is sufficient to cause actuation of a switch 234 suitably mounted in chamber 186. At the same time, the initial movement of the brake pedal and the resultant increase in fluid pressure in the system, causes hydraulic switch 18 (FIG. 2) to become closed. Hydraulic switch 18 is fully disclosed and described in copending U.S. patent application Ser. No. 203,075, filed June 18, 1962. Basically, switch 18 comprises piston 236 which has connected thereto at its upper end a metallic contact cup 238. Cup 238 is adapted to bridge contacts 240 and 242 but is normally maintained spaced therefrom by spring 244. An adjusting screw 246 is provided by means of which the tension of spring 244 may be varied, thus enabling the switch 18 to be preadjusted whereby a predetermined fluid pressure, e.g. 120–125 p.s.i., will cause piston 236 to move against the action of spring 244 sufficiently to bridge the contacts 240, 242. Actuation of switch 18 in the aforedescribed manner enables current to flow from battery 248 through contact 242, contact cup 238, and contact 240 to line 250 to the solenoid coil positive terminal 252 (FIG. 4). The voltage is applied through the solenoid coil 254 and then through line 256 to terminal 258 on switch 234. When switch 234 is open, the circuit to ground is not complete, and solenoid 188 will not be energized. However, upon actuation of switch 234 in the manner aforedescribed, the circuit to ground is completed through terminal 260, line 262 and ground connection 264. It is important to note that the circuit is such that although it is essential for hydraulic pressure switch 18 to be closed in order for solenoid 188 to be energized, closing of switch 18 alone is not sufficient, but rather switch 234 must also be actuated in order to complete the circuit. This is important because under normal operating conditions of the system sufficient pressure will normally be introduced to switch 18 by the foot pedal to actuate same, but it is essential that solenoid 188 not be energized under such normal conditions.

Energization of solenoid 188 accomplished by rightward movement of the cam gear (FIG. 5) closing switch 234, causes plunger 190, 192 to be retracted against the action of spring 194, thus freeing the shaft 200 and spool 220 for axial movement under the influence of the governor assembly. Therefore, the cam gear operating in response to the inertia of the governor and the solenoid operated plunger together can be regarded as an inertia-responsive means for initiating the action of the governor.

As soon as the solenoid plunger has been retracted, due to the sudden slowing at the commencement of a skid condition, the governor assembly takes over control and, as it is stopped by stop surface 225 of the cam gear, spring 218 will cause the governor assembly to commence to collapse, thus moving shaft 200 and spool 220 axially to the left, when viewing FIG. 4. This movement of spool 220 causes corresponding rotary movement of valve 126, thus shutting off the passage of atmosphere to brake control conduit 104 and simultaneously effecting communication between vacuum passage 119 and conduit 104. This results in suction being introduced to chamber 98 which in turn overcomes the action of spring 100 to cause pistons 84 and 68 to move to the left, viewing FIG. 2. This movement of piston 68 enables ball valve 64 to close due to the action of spring 62, thus preventing the introduction of further fluid pressure to the system via brake pedal 12. Further leftward movement of piston 68 enables the fluid in lines 78 and 80 to back up, thereby relieving the pressure until the vehicle brakes begin turning, the duration that the suction is applied, and hence the degree of vacuum and the amount of pressure relief depending on the slipperiness of the road. As soon as the brake release pressure is achieved, the vehicle wheels commence to rotate, whereby drive shaft 142 through the aforedescribed gearing once again causes gears 170 and 222 to rotate, it being apparent that this renewed rotation of shafts 200 and 202 will cause the governor assembly to once again turn, and due to the centrifugal force created, to retract shaft 200 and spool 220 to once again rotate valve 126 to the positions illustrated in FIGS. 2 and 4, where once again atmosphere is connected to chamber 98, thus enabling brake pressure to be restored.

But, it is very important for my present invention that the solenoid is not de-energized and the governor returned to the restraint of the inertia-responsive device. Rather, the governor remains free to act. It leaves the valve 126 open to atmosphere 118 for an instant until the degree of vacuum is lessened enough to cause slight rightward movement of the piston 84 to cause the brake pressure to slightly exceed release pressure and a skid again commences. Again the governor is slowed by the drive train and again it opens the valve 126 to suction 116, and this cycle repeats again and again in an extremely rapid hunting or modulating action with the gas pressure in the conduit 104 never reaching either atmospheric or engine vacuum level, but rather modulates at intermediate levels. In dependent fashion, the hydraulic brake pressure modulates narrowly about the release pressure as a mean, without returning to normal pressure. When the skid condition ends, the governor opens the valve continually to atmosphere and pistons 84 and 68 move fully to the right to once again uncoat isolation valve 64 and return normal brake operation. The faster the vehicle travels, the faster does this action occur, but the action is sufficiently fast down to speeds on the order of 10 m.p.h. and even lower, to rapidly pump and release the brakes. It is another very

[PX 13, p 13]

important aspect of the invention that the governor relinquishes control of the brake pressure before the vehicle stops.

In order to effectively carry out this modulating action, I have found it to be critical, in the preferred construction, that the solenoid 188 remain energized or that the plunger in some other way remains retracted until the vehicle slows to a predetermined slow speed, e.g. 5 or 10 m.p.h., depending on the vehicle. Since closing of switch 234 to complete the circuit to ground that energizes solenoid 188 is only momentary, holding means are provided for maintaining the solenoid energized until hydraulic pressure switch 18 opens, it being understood that switch 18 is preset to open only when fluid pressure in the system falls below the minimum value possible to achieve a skid condition, e.g. set 85–90 p.s.i. produced by the governor at 5 or 10 m.p.h.

The aforesaid holding means for the solenoid comprise a terminal 266 having a connection to ground as at 268, which terminal is engaged by plunger 190 when the latter is retracted in order to complete the circuit to ground independent of switch 234. Thus, although closing of switches 18 and 234 is essential in order to energize solenoid 188, once the solenoid has been energized and its plunger retracted, switch 234 may be opened without de-energizing the solenoid, and the solenoid will remain energized until switch 18 opens.

As hereinbefore stated, it will be obvious that if switch 18 were not utilized, then there would be no means present for insuring de-energization of the solenoid once the condition has been reached where a skid is no longer possible, it being obvious that if the solenoid were still energized at such low speeds, the vehicle would be without effective brakes, since the governor assembly would not exert sufficient centrifugal force to overcome the action of spring 213 and move shaft 200 and spool 220 to the rightward position (FIG. 4) wherein valve 126 enables the brakes to be controlled by pedal 12.

It is possible for the reset device to take some other form, such as a pendulum device or a vehicle inertia memory system that can detect the slow speed of the vehicle, but brake pressure-sensitive devices, and particularly hydraulic-electric switches have important advantages in the system, particularly since they can respond whenever the skid condition has passed, regardless of the speed of the car, to enable normal operation, while offering low speed protection against the governor's cutting off the brakes completely. The adjustment of the pressure level to which the device responds can be adjusted according to the wishes of the operator and the type of road conditions to be expected. But, in fact, a single setting, below that at which skidding on ice occurs, which is determinable for any given vehicle, is advantageously employed to give anti-skid protection under virtually all conditions.

Any tendency for the unit 24 to erroneously sense a skid condition and actuate switch 234 in response to a rapid deceleration is overcome by use of properly calibrated weights 208 and a properly tensioned spring 230. In addition, I have found that an erroneous sensing is most likely to occur where there is a rapid acceleration of the drive shaft quickly followed by a rapid deceleration. The frictional brake or drag that is resiliently exerted on spool 220 by plunger 190 helps to overcome this problem since this action tends to reduce the torque applied to shafts 200 and 202 by the governor assembly. It has been found, however, that at high speeds the frictional engagement between plunger 190 and spool 220 results in undue wear of the parts. Thereof, plunger 190 is assisted by spring 230, and the plunger is disengaged at high speeds. To this end, stop means 198 engages flange 196 so as to limit the outward movement of plunger 190 whereupon no engagement exists between plunger 190 and spool 220 at relatively high speeds. I have realized that the crucial problem of rapid acceleration followed by rapid deceleration occurs at

low speeds in the low gear range with conventional automobiles, and I have, therefore, found that it is not necessary to have plunger 190 exert a frictional drag on spool 220 and shaft 200 at relatively high speeds.

Although the operation of my invention is thought to be clear from the foregoing description, a brief resume of the operation will now be given. Under normal operating conditions, solenoid 188 is de-energized whereby the spring-loaded solenoid plunger maintains the skid-sensing and control unit inoperative. The de-energized solenoid further insures that rotary valve 126 is in the position wherein atmosphere is introduced to chamber 98. Since chamber 96 is also open to atmosphere, it follows that there is a pressure balance on each side of large piston 84 thus enabling spring 100 to take over and move the piston 84 to the position illustrated in FIG. 2. In this position, piston 68, which is connected to piston 84 and moves therewith, engages ball valve 64 so as to unseat same against the action of spring 62. With ball valve 64 so opened, fluid pumped from master cylinder 14 by brake pedal 12 is free to pass directly to hydraulic lines 78 and 80 to actuate the vehicle brakes. The pressure-sensitive switch is adapted to be closed by a low level of brake pressure, e.g. 120–125 p.s.i., placing the anti-skid unit into readiness for operation.

Upon encountering a skid condition, i.e., a condition wherein the vehicle operator applies more brake pressure than is required for a particular road surface, the vehicle wheels slow suddenly. This causes corresponding deceleration of drive shaft 142, and through suitable gearing, drive gear 170 and cam gear 222 also slow abruptly. The inertia of the governor assembly causes the gear 222 to move longitudinally by the cam surface aforedescribed, said longitudinal movement of gear 222 causing switch 234 to be closed which results in energization of solenoid 188 and the retraction of solenoid plunger 190. The second switch or terminal 266 is engaged upon retraction of the solenoid plunger to maintain the solenoid energized until switch 18 opens, even though switch 234 has now opened. Switch 18 is preset, e.g. at 85–90 p.s.i. to be opened at a brake pressure below that at which skids can occur, e.g. 100–150 p.s.i. for ice, whereupon solenoid 188 will remain energized until possibility of skidding no longer exists, and the solenoid is then de-energized to restore normal foot brake action before the governor has a chance ever to totally relieve the brakes.

Regarding the operation of the governor, once the solenoid 188 has been energized, the governor control unit takes over. More specifically, when a skid condition is approached, the sudden slowing of the vehicle wheels causes the governor assembly to collapse under the action of spring 218, thus moving shaft 200 and spool 220 to rotate valve 126, this movement now being permitted due to retraction of plunger 190. Rotation of valve 126 blocks the flow of atmosphere to chamber 98 and simultaneously allows the introduction of a vacuum to said chamber. The introduction of vacuum overcomes spring 100, thus resulting in movement of pistons 84 and 68 to the left, viewing FIG. 2. This movement closes isolation valve 64, thus preventing the application of further fluid pressure by brake pedal 12. As the valve remains in this position, the degree of vacuum increases, causing further leftward movement of piston 68 (FIG. 3), so that pressure in lines 78 and 80 and the brakes are relieved. A point is reached when the brakes are relieved sufficiently to allow the wheels to speed up from their slowed condition, the degree of relief required depending on the degree of slipperiness of the road. Then shafts 200 and 202 speed up once again, whereby the governor assembly retracts shaft 200 and spool 220 to once again rotate valve 126 to the position where atmosphere is introduced to chamber 98 and the piston 68 moves to the right and increases the brake pressure, until a part is reached where the wheels slow again and the cycle repeats. This action continues in a hunting manner so long as

[PX 13, p 13]

the vehicle is in an incipient skid, thus insuring that maximum brake pressure is continually applied to the wheel brakes short of pressure that would cause locking of the wheels. Once the brake pedal is released or the brake pressure in the system has dropped below the minimum possible value to create a skid, hydraulic pressure switch 18 breaks the circuit to solenoid 188, whereupon the latter becomes de-energized and the brakes once again become under complete control of the operator.

It is important to note that although loss of vacuum in the vehicle system will render the anti-skid means ineffective, said loss of vacuum will in no way affect normal operation of the vehicle brakes. Also, it is important to note that this system can be used with foot pedal controlled brakes of both the powered and nonpowered types.

While a spring-loaded fly ball type of governor has been illustrated, and is presently preferred, the broad concept of the invention include the use of other governors, all rotary weight governors, such as gravity loaded fly ball, centrifugal shaft and inertial shaft governors, as well as governors that operate through electrical action.

The concepts are likewise applicable to other brake systems including air brakes in which valve 126 could directly modulate air brake pressure. Likewise it is possible to separate the governor from the valve, let the governor actuate a switch and let the valve be actuated by a solenoid responsive to the condition of the switch.

While use wth engine vacuum is very efficient and inexpensive, and unlike previous systems I have proposed, does not cause the engine to run roughly due to loss of vacuum, the invention is applicable to other sources of pressure differential, incompressible fluid as well as gaseous fluid and numerous aspects of the invention are useful with nonfluid types of brake systems.

Referring now to FIGS. 7 and 8, a slightly different form of my invention is disclosed in which the relieving device is incorporated as an integral part of a power brake unit. The system generally comprises a brake pedal 300, a master cylinder 302, an accumulator 304, a safety valve 306, a hydraulic pressure switch 308, a power brake control unit 310, and an anti-skid unit 312.

Actuation of brake pedal 300 operates piston 314 to pump fluid from cylinder 316 to conduit 315. Master cylinder 302 further comprises a fluid reservoir 320 communicating with cylinder 316 by means of passageway 322. A conduit 324 extends from reservoir 320, the function of which will hereinafter be described. Accumulator 304 is in communication with conduit 318 and comprises a spring loaded piston 326 and acts as a cushion and shock absorber for the fluid system where variations in fluid pressure occur.

During normal operation of the vehicle with which the power brake unit is associated, and with the brakes in their non-actuated position, the power brake unit 310 is in the position illustrated in FIG. 8. In this position, chambers 328 and 330 of cylinder 332 are both under vacuum, thereby equalizing the pressure on opposite sides of piston 334. This enables spring 336 to urge the piston 334 to the leftward position illustrated, said piston 334 being connected by means of rod 338 to a brake actuating piston 340. Piston 340 is slidably mounted in cylinder 342, said cylinder having an outlet conduit 344 leading to safety valve 306. Conduit 344 is also in communication with hydraulic pressure switch 308.

A supply of hydraulic fluid is located in cylinder 342 whereupon movement of piston 334 against the action of spring 336, and corresponding movement of piston 340 will force fluid from cylinder 342 under pressure through conduit 344 and with valve 306 in the position illustrated in FIG. 8 to conduit 346 and then to hydraulic lines 348 which lead to the wheel cylinders (not shown) of the vehicle's braking system. Thus it will be seen that movement of piston 334 and 340 to the right, viewing FIG. 8, will cause operation of the vehicle brakes.

As hereinbefore indicated, under normal operating conditions of the vehicle, and with the brakes not applied, pistons 334 and 340 are in the position illustrated in FIG. 8 due to the fact that a vacuum exists in both chambers 328 and 330. More specifically, suction from the intake manifold 360 of the vehicle passes through conduit 352 into chamber 330 and then through passageway 354, through openings 356 in diaphragm 358 to conduit 360, through rotary valve 362 of the anti-skid unit 312, through conduit 364 and then to chamber 328. The control unit generally shown at 366 which forms a part of power brake unit 310 is of conventional form and comprises the aforesaid diaphragm 358, hydraulically movable by means of piston 368 to a closed position wherein the diaphragm engages the fixed seat 370 to block communication between passage 354 and conduit 360. The piston 368 also controls movement of poppet valve 372 normally maintained closed by spring 374. Spring 376 normally urges diaphragm 358 to its open or unseated position, as illustrated in FIG. 8. An atmosphere intake is located at 378, and introduction of atmosphere into the system is controlled by valve 372 in a manner now to be described.

As above indicated, with the parts in the positions illustrated in FIG. 8, the brakes of the vehicle are nonoperative due to the fact that suction is free to circulate through conduit 352 to chamber 330 and then through control unit 366 to conduit 360, through rotary valve 362 to conduit 364 and then to chamber 328. With vacuum on both sides of piston 334, spring 336 maintains the piston 334 and the connected brake actuating piston 340 in brake release position. Upon actuation of brake pedal 300 by the operator, pressurised fluid passes through conduit 313 to conduit 380. With the safety valve 306 in the position illustrated, flow of the fluid into the valve is blocked, and hence the brake fluid is forced to flow through conduit 380 into the brake control unit 366. None of the fluid from the pedal is allowed to reach the brake actuators at 348.

Introduction of the pressurized fluid to the brake control unit 366 causes actuation of piston 368 to move diaphragm 358 to its closed or seated position and to simultaneously open poppet valve 372 in a conventional manner. This movement blocks communication between chamber 330 and conduit 360 and simultaneously allows air to enter through inlet 378 through the now open valve 372 to conduit 360. The atmosphere circulates around conduit 360 through valve 362 to conduit 364 and then the chamber 328. The introduction of atmospheric pressure to chamber 328 creates a pressure differential which overcomes spring 336 and moves piston 334 and piston 340 to brake actuating position. More specifically, the movement of piston 340 in cylinder 342 forces the brake fluid through conduit 344 and safety valve 306 to conduit 346 and hydraulic brake lines 348 to actuate the vehicle brakes in known fashion.

In order to insure that there is always an adequate supply of brake fluid in cylinder 342, conduit 324 connects fluid reservoir 320 to the cylinder 342, it being important to note that the inlet 382 to the cylinder 342 is located behind the forward portion of piston 340. Piston 340 is provided with a one-way valve 384 which enables fluid to pass through the piston from left to right, viewing FIG. 8, but which blocks flow of fluid through the piston from right to left. Thus, during operation of the piston 340, the suction created thereby in cylinder 342 sucks fluid through conduit 324 from reservoir 320 into cylinder 342 on the rear side of piston 340. During the return stroke of piston 340, the valve 384 allows fluid to pass through the piston, thereby insuring that there is always a full supply of fluid in the cylinder for brake actuation.

Since the structure and operation of hydraulic pressure switch 308 and anti-skid unit 312 are identical to that already described in connection with the first form of my invention, it is not thought that further description is

necessary. It will be noted, however, that where the operator applies more brake pressure than is required for a particular road surfaces, the vehicle wheels will lock, whereby the same structure and operational sequence will cause rotary valve 362 to be moved to a position wherein conduit 360 becomes blocked and conduit 352A becomes open to allow suction from manifold 350 to be introduced through conduit 364 to chamber 328, whereupon spring 336 will once again take over to release the brakes. The same modulating action as aforedescribed will take place until hydraulic pressure switch 303 eventually renders the anti-skid inoperative.

In accordance with my invention it is crucial that the entire brake pressure be controlled by the anti-skid unit and that no fluid be conducted directly from the foot pedal to the wheels. Means are provided to insure that brake pressure will not be lost should there be a loss of suction. This is accomplished by safety valve 306. More specifically, conduit 352D is connected to the suction conduit 352 whereupon as long as proper suction exists in the system, piston 386 is maintained in the position illustrated in FIG. 8 against the action of spring 388. In this position, conduit 344 is in communication with conduit 346, but conduit 350 is blocked at its extremity adjacent the safety valve. Should suction be lost in the system, spring 388 will become effective to move piston 386 upwardly, thereby blocking the communication between conduits 344 and 346 and at the same time providing communication between conduit 320 and conduit 346. With the safety valve in this position, the brakes are directly controlled by operation of pedal 300 since the entire power unit is now bypassed.

A one-way check valve 390 is located in conduit 352 adjacent the intake manifold 350 since without same backfire of the vehicle engine would cause an explosive pressure to be introduced to the power unit, which would be undesirable for obvious reasons.

Referring once again to FIG. 4, it will be noted that should the vehicle move in reverse, the direction of rotation of drive shaft 142 will likewise be in reverse, thus causing reverse rotation of flexible shaft 164 and shaft 168. Reverse rotation of shaft 168 will, in turn, cause reverse rotation of drive gear 170 and cam gear 222, causing pin 228 to leave stop surface 227 and move toward stop surface 225, which, as previously described, causes gear 222 to slide on shaft 202 in a direction away from partition 180 to close switch 234 and thus activate the governor control and anti-skid means. Even though it is possible that very slow reverse movement of the vehicle might not be sufficient to cause movement of cam gear 222 to activate the governor control, the fact remains that there is always the possibility and even likelihood that reverse movement of the vehicle will activate the governor control even though no skid condition exists. Since there is no practical need for any anti-skid means when the vehicle is moving in reverse, and since the activation of these means when not needed might even result in the loss of some stopping distance, it has been found desirable to provide means for rendering the governor control and anti-skid ineffective and inoperative when the vehicle moves in reverse.

Referring to FIGS. 9 through 12, a somewhat modified anti-skid unit is shown generally at 400. Unit 400 is basically the same as skid-sensing and control means 24 hereinbefore described except that a reverse cut-off switch has been incorporated in the unit to prevent energization of solenoid 188 when the vehicle drive shaft is in reverse and flexible shaft 164 correspondingly rotates in a reverse direction. The reverse cut-off switch, shown generally at 402 (FIG. 10) comprises a shaft 404 slidably coupled to shaft 164 as at 406, in the same manner that shaft 168 was drivingly engaged to flexible shaft 164. Shaft 404, which is constructed of any electrically conductive metal, is journalled in bore 408, said bore having an enlarged end portion 410, it being understood that the larger diameter of portion 410 results in the presence of an annular shoulder 412 at the extremity of bore 404. Slidably mounted in portion 410 is a contact button 414, said button being resiliently urged toward bore 404 by means of spring 416. The opposite end of spring 416 resiliently engages a contact cup 418 having a terminal 420 extending radially therefrom.

Adjacent its opposite extremity, shaft 404 has fixedly mounted thereon the afore-described drive gear 170, said gear meshing with cam gear 222, all in the manner hereinbefore described. A spring 422 engages washer 424 to normally urge shaft 404 to the left when viewing FIGS. 9 and 10, thus causing the end of shaft 404 to extend into enlarged portion 410 and, therefore, into engagement with contact button 414, as illustrated in FIG. 9. Loosely mounted on shaft 404 is a reverse switch cam 426 comprising a circular plate 428 having a spiral cam portion 430 mounted thereon, and further having an outwardly extending pin 432 located adjacent the end of cam track 430. A cam pin 434 extends radially from shaft 404 and is adapted to cooperate with the spiral cam track 430 to determine the axial position of shaft 404. Expressed differently, when shaft 404 is rotating in a counterclockwise direction when viewing FIG. 12, cam pin 434 will engage the stop pin 432 at the low point on the spiral track 430. With the parts in this position, shaft 404 will assume the position illustrated in FIG. 9 wherein the extremity of the shaft is in engagement with contact button 412, it being understood that spring 422 is somewhat stronger than spring 416, whereupon the extremity of shaft 404 will actually enter into enlarged portion 410. Upon clockwise rotation of shaft 404, pin 434 will ride up the spiral cam track 430 until it reaches the high end of said track and engages the opposite side of stop pin 432, as illustrated in FIG. 10. In this position, shaft 404 has been moved to the right (when viewing FIG. 10) against the action of spring 422 sufficiently for the end of said shaft to become completely withdrawn from enlarged portion 410. Since the annular shoulder 412 prevents the contact button 414 from moving out of enlarged portion 410, it follows that the end of shaft 404 becomes disengaged from contact button 414, as clearly illustrated in FIG. 10. Also mounted on shaft 404 is a washer 436 having a radially extending contact arm 438, the purpose of which will hereinafter be described. As will be seen most clearly in FIG. 10, the washer 436 is mounted on shaft 404 adjacent the flat rear surface of cam 426, it being understood that shaft 404 may freely rotate within washer 436. A cam thrust washer 440 comprising a peripheral ring portion 442 and bent marginal lugs 444 resiliently engages the opposite face 428 of cam 426, whereupon to resiliently urge cam 426 into engagement with washer 436 and at the same time to impart a sufficient frictional drag on cam 426 so that movement of shaft 404 will not impart corresponding rotary movement to the cam until pin 434 moves into engagement with stop pin 432. The thrust washer 440 has a radially extending spring arm 446 that is secured to the housing 176 by any suitable means, such as by having the end of the spring arm clamped between adjacent portions of the housing, as illustrated in FIGS. 9 and 10.

The operation of reverse switch 402 is as follows. Assuming that shaft 404 is being rotated in a counterclockwise direction, which for purposes of this description corresponds to forward movement of the vehicle, the switch will, as aforestated, assume the position illustrated in FIG. 9 wherein shaft 404 is making engagement with contact button 412. With the parts in this position, and assuming that a skid condition commences to exist, causing cam gear 222 to close switch 234, current will flow through line 262 to terminal 420 through contact cup 418, spring 416, contact button 414, and then to shaft 404, which as previously described, is in engagement with contact button 414. Shaft 404 is in turn in electrical engagement with washer 436 and its radially extending ter-

minal 438, both directly and through the engagement of pin 434, which in turn engages cam 426, which in turn bears against washer 436. Thus, the flow of current is through terminal arm 438 and then to line 262A, which connects to ground as at 264. This, along with line 256, completes the circuit to solenoid 188 in order to energize same and render the governor control effective, all in a manner hereinbefore described. On the other hand, should the vehicle move into reverse, thus resulting in clockwise rotation of shaft 404, cam 426 will cause the shaft 404 to retract and become disengaged from contact button 414, as illustrated in FIG. 10, thus interrupting the connection to ground and preventing energization of solenoid 188. It will thus be seen that whenever the vehicle and its drive shaft moves in reverse, the reverse cut-off switch 402 will automatically prevent the skid sensing and control means and the governor control from coming into play. Thus, the vehicle brakes will always be under normal and direct control by the operator whenever the vehicle is moving in reverse.

While there is shown and described herein certain specific structure embodying the invention, it will be manifest to those skilled in the art that various modifications and rearrangements of the parts may be made without departing from the spirit and scope of the underlying inventive concept and that the same is not limited to the particular forms herein shown and described.

What is claimed is:

1. Acceleration responsive means for a vehicle wheel anti-skid system in a vehicle brake system having a fluid source, a vehicle wheel brake operating actuator cylinder in fluid communication with said fluid source and operator actuated means for applying fluid pressure to said vehicle wheel brake operating actuator cylinder, said vehicle wheel anti-skid system comprising said acceleration responsive means which senses a predetermined deceleration of the vehicle wheel, means actuated by said acceleration responsive means for alternately applying and relieving the fluid pressure applied to the vehicle wheel brake operating actuator cylinder when said predetermined deceleration of the vehicle wheel is sensed by said acceleration responsive means, and means for controlling said means for alternately applying and relieving the fluid pressure, said acceleration responsive means comprising a governor assembly, a rotatable shaft operatively connected to said governor assembly to rotate the governor assembly, a cam gear mounted on said shaft, means for operatively connecting said cam gear to said shaft so that said shaft may be driven in rotation by said cam gear when said cam gear is rotated, said cam gear being mounted for sliding movement endwise of said shaft, means for rotating said cam gear dependently with the speed of rotation of the vehicle wheel, means for sliding said cam gear endwise of said shaft upon a predetermined deceleration of the vehicle wheel, said cam gear being capable of actuating said control means, whereby said cam gear normally rotates said shaft which in turn rotates said governor assembly but upon attainment of a predetermined deceleration of the wheel said cam gear is moved endwise of said shaft temporarily disconnecting the driving relationship between the cam gear and the shaft and actuating said control means so that the fluid pressure applied to the vehicle wheel brake operating cylinder will be alternately applied and relieved.

2. Acceleration responsive means as set forth in claim 1 wherein said means for operatively connecting said cam gear to said shaft includes a pin secured to and extending radially of said shaft and means formed in said cam gear to engage said pin.

3. Acceleration responsive means as set forth in claim 2 wherein said means formed in said cam gear includes a recess in one end of said cam gear, a helical surface formed in said cam gear within the confines of said recess, a first stop surface at one end of said helical surface and a second stop surface at the other end of said helical surface, said pin being adapted to fit within said recess and contact said helical surface.

4. Acceleration responsive means as set forth in claim 3 including a spring finger mounted to bias said cam gear into engagement with said pin so that when said cam gear is rotated said pin engages said first stop surface thereby to rotate said shaft but upon attainment of the predetermined deceleration of the wheel said pin rotates relatively to said cam gear thereby to slide along said helical surface until said pin abuts said second surface thereby to slide said cam gear endwise of said shaft until the cam gear engages and actuates said control means.

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DUANE A. REGER, Primary Examiner

U.S. Cl. X.R.

188—181

[PX 14, p 1]

PLAINTIFF'S EXHIBIT 14

SINGER COMPANY OF CANADA LTD.
ST. JOHNS PLANT — 200 ST. LOUIS STREET — ST. JOHNS, QUE.

January 21st, 1964.

Mr. Frank A. Perrino,
Perma Research & Development Co.,
345E Washington St.,
North Attleboro, Mass.,
U.S.A.

REC'D JAN 23 1964

Dear Frank:

Further to our telephone conversation today, I am sending along the name and address of the person to contact at our Elizabeth Plant regarding possibility of Singer producing your Anti-Skid device in the States.

Mr. Robert A. Kloby,
Director,
Forward Planning,
Elizabeth Plant,
The Singer Company,
321 First Street,
Elizabeth, N.J.

Mr. Kloby had a brief talk with George Mercier this week, and he expects to hear from you to arrange a meeting.

Yours very truly,

SINGER COMPANY OF CANADA LIMITED

Keith H. Heffer,
Sales Manager.

KHH/ct

cc: Mr. Robert A. Kloby

[PX 15, p 1]

PLAINTIFF'S EXHIBIT 15

January 24, 1964

Mr. Robert A. Kloby
Director of Forward Planning
The Singer Plant
321 First Street
Elizabeth, New Jersey

Dear Mr. Kloby,

On January 21, 1964, we received a letter from Mr. Heffer of your organization and he suggested we write to you to arrange a meeting.

We will be pleased to meet with you at your convience to discuss the manufacturing phase of the Perma Anti-Skid Control.

You are cordially invited to visit our facilities in order that you may meet our staff and also to drive one of our test cars equipped with a Perma Anti-Skid Control.

Please call Mr. Krawczyk or myself or write at your earliest convience so that we may arrange a meeting.

Very truly,

Frank A. Perrino
President

cc: Keith H. Heffer
George Mercier

FAP/ms

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[PX 17, p 1]

PLAINTIFF'S EXHIBIT 17

THE SINGER COMPANY

ELIZABETH PLANT • 321 FIRST STREET, ELIZABETH, NEW JERSEY • 201 EL 2-8200

RECT FEB 21 1964

February 19, 1964

Mr. F.A. Porrino, President
Perma Research and Development Company
345 E. Washington Street
North Attleboro, Massachusetts

Dear Mr. Porrino:

Further to my letter of February 17th and confirming my conversation with Mr. Krawczyk, we shall proceed with the value engineering and pricing of the Perma Anti-Skid Control Unit as soon as we receive the necessary information from you.

I have requested Mr. Krawczyk to forward the following:

1. Sepias of all prints, including assembly prints.
2. Three anti-skid units.
3. Quality specifications, testing specifications and any other necessary specifications that do not appear on the prints.
4. List of vendors who are now supplying components.
5. Your best estimate of monthly requirements or individual order requirements, taking into consideration possible seasonal fluctuation. This will be required for a determination of the number of setups.
6. A bill of materials.
7. A cutaway model of the unit, if one is available.

I did not request the last two items in my conversation with Mr. Krawczyk but am doing so now because they will be most helpful in our analysis.

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[PX 17, p 2]

We shall have a price developed in about 30 days after receipt of the aforementioned material. Also, at that time we shall be in a position to estimate dates for production samples and production parts.

Thank you very much for your cooperation, and I shall be in contact with you as this project progresses.

Yours very truly,

The Singer Company

Robert A. Kloby
Director of Forward Planning

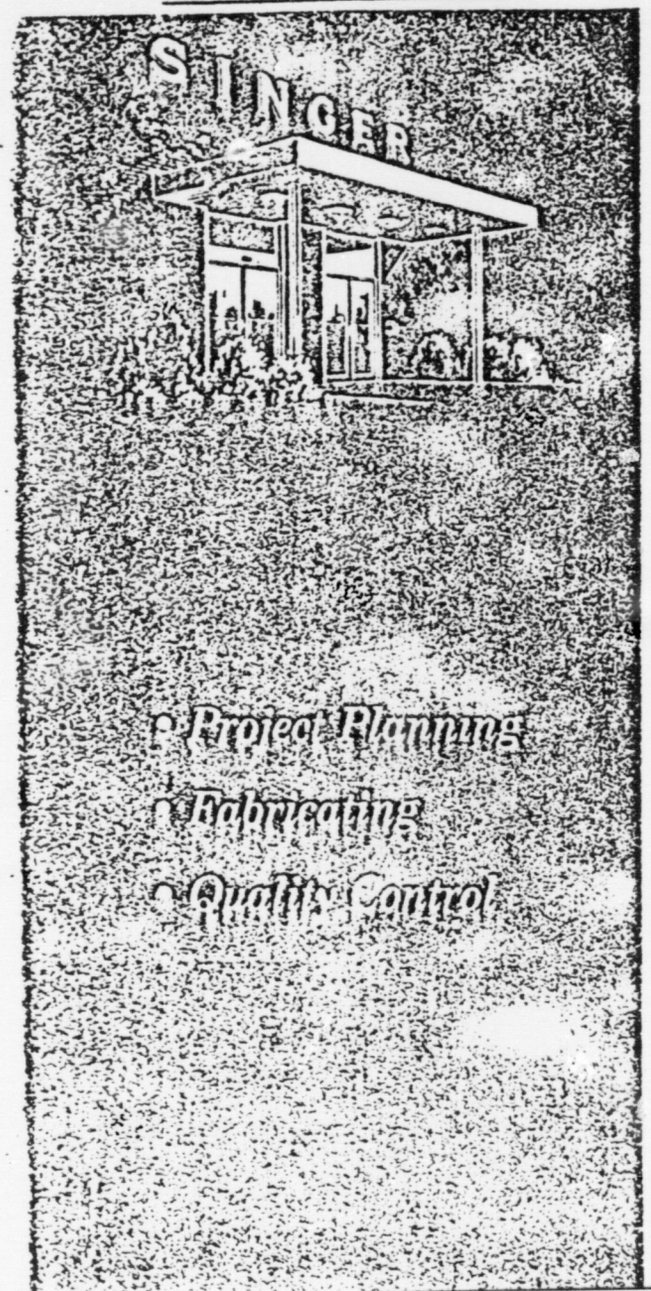
RAK:hl

[PX 13, p 7]

E-91

[PX 18, p 1]

PLAINTIFF'S EXHIBIT 18



SINGER-ELIZABETH
DIVISION
MANUFACTURING SERVICES

THE SINGER MANUFACTURING COMPANY

[PX 18, p 1]

E-105

Quality Control

Quality is not accidental. It must be planned and built into a product.

Quality depends on people. The people in this supporting facility, ever on the alert, demand and achieve the quality synonymous with SINGER.

All inspection controls are based upon current statistical data and techniques. A Quality Control Manual has been issued by this department to formalize procedures, methods and instructions. It is used as a guide in performing quality control functions. Military Standard #105-C, Sampling Procedures and Tables for Inspection by Attributes, is also used.

A unique feature of the Quality Control Department is its ability to detect a defect early in production—perhaps as early as the engineering and laboratory phases of planning. Such an accomplishment can eliminate a staggering amount of scrap and re-work.

This team of workers, set-up men, supervisors, engineers and inspectors acts unfailingly to effect quality improvements and cost reduction, providing end products that satisfy customer requirements.

Along with the Metallurgical Laboratory and the Engineering staff, the SINGER-ELIZABETH Quality Control Department upholds the sound ideals of top-quality and low costs which make SINGER products praised and desired the world over.

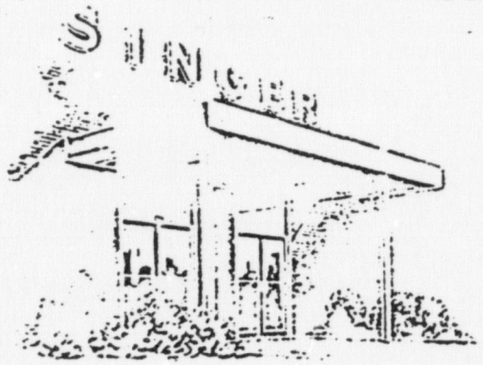
grow with **SINGER-ELIZABETH**



[PX 18, p 2]

[PX 18, p 2]

PLAINTIFF'S EXHIBIT 18A



[PX 18A, p 1]

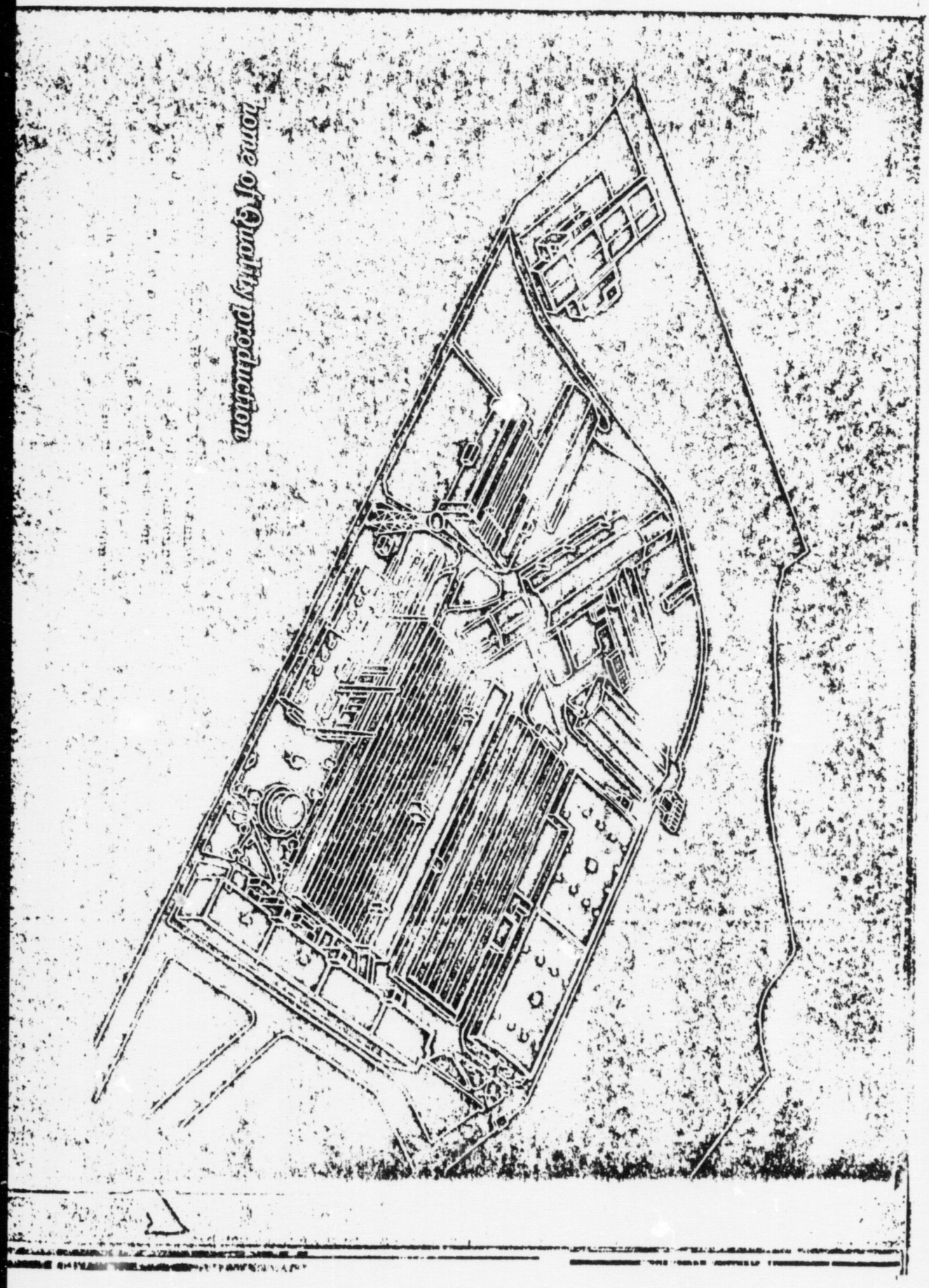
SINGER-ELIZABETH DIVISION

MANUFACTURING SERVICES

[PX 18A, p 1]

E-107

THE SINGER MANUFACTURING COMPANY


Zone of Quality Production

[PX 18A, p 3]

THE SINGER MANUFACTURING COMPANY is known throughout the world as the leading producer of sewing machines and related products.

For over a century it has constantly improved and expanded its household and industrial lines to accommodate the entire sewing and textile trade.

A very important fact, not as well known, is that the comprehensive facilities of this company enable it to design, mass-produce and distribute many products essential to industries other than those in the textile trade.

A continuing program of increasing the efficiency of SINGER-ELIZABETH permits both the expansion of its sewing machine production and the release of a large portion of its productive capacity to a general manufacturing service program. The surplus capacities are thus made available to you.

Forgings, metal stampings, gray-iron castings, aluminum and zinc die castings, machined parts, tools, gauges, dies and other precision-made parts are everyday products of the Elizabeth, New Jersey plant.

Assembly and sub-assembly of small, intricate parts, as well as large, complete units are accomplished with the knowledge and skill of many years of experience.

Since these metal working and metal finishing operations are available here at one site, low costs are assured. Located on main arteries for rail and truck, with air freight terminals but minutes away, SINGER-ELIZABETH can accomplish rapid deliveries.

Each progressive step in the creation of a particular product is firmly controlled by a guidance and inspection service designed to honor the highest recognized standards of quality and performance. However, this supporting facility does share with every other department at SINGER-ELIZABETH in an honest respect for deadlines.

SINGER-ELIZABETH believes that its services can be of help to you—the Manufacturer—can mean the difference between meeting a delivery date and costly delays. SINGER-ELIZABETH believes that they can increase your production possibilities far beyond your present capacity to do so alone.

Ask SINGER-ELIZABETH to talk over your next problem with you—let them help you solve it quickly and completely to your satisfaction.

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Foundry

Die Casting

Power Press and Forge

Hardening and Annealing

Polishing and Plating

Tools and Gauges

Screw Machines

Assembly

Electrostatic Painting

E-109

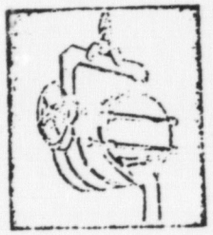
[PX 18A, p 3]

[PX 18A, p 4]



high quality castings
63 heats - 45 tons daily

Foundry



Fountainhead of gray-iron metal working operations, the Foundry is one of the most modern and fully equipped facilities of its kind in the area.

The SINGER-ELIZABETH Foundry, recently modernized when it moved into new quarters that cove. 46,620 square feet, can produce high quality castings ranging in weight from ounces to one hundred pounds. Within this range, this department is ready and able to fill your cast iron requirements economically—to fit your specifications and your schedule.

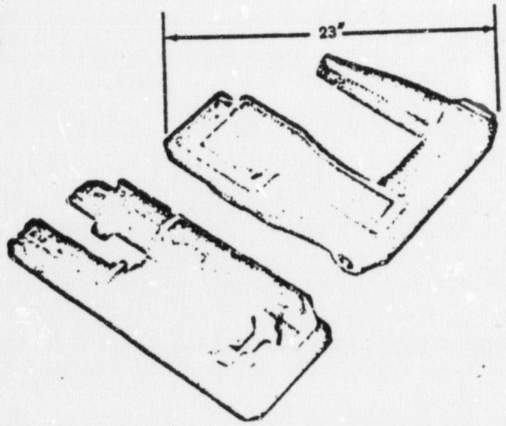
From automatic moisture-cor rolled sar.d handling to thorough polishing, the entire operation of the Foundry is controlled for full round-the-clock capacity with maximum quality and minimum waste. Every step, from selection of sand for molds and determination of chemical composition of molten metal to final inspection, is safeguarded through the continuous consulting services of SORBO-MAT Process Engineers.

E-110

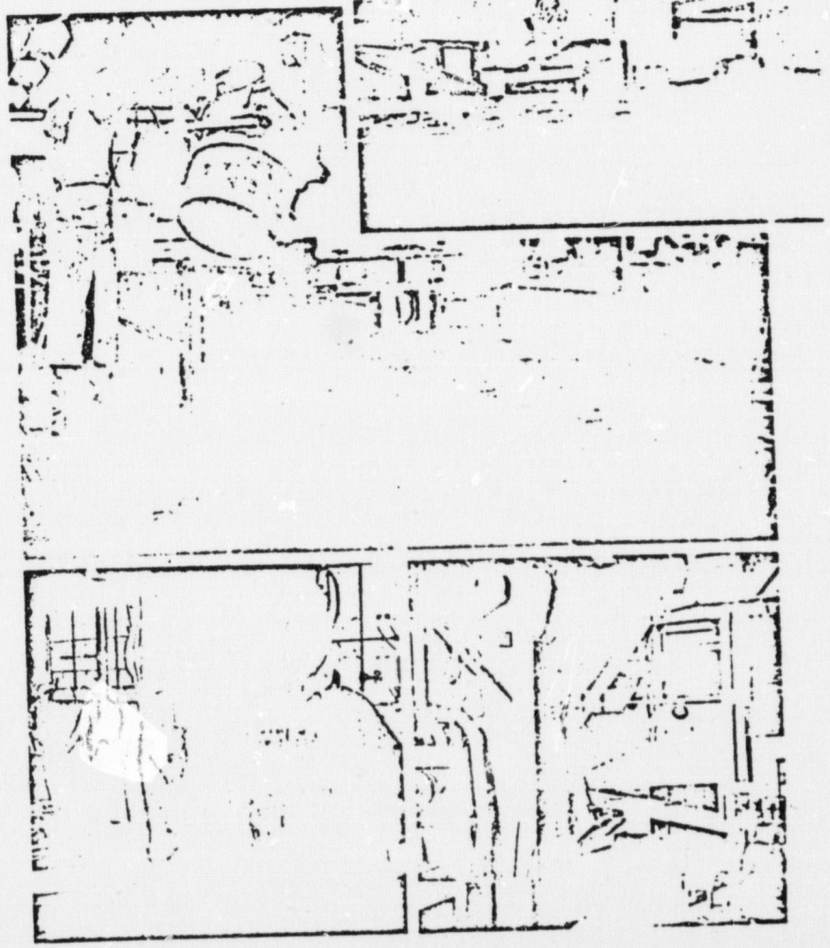
[PX 18A, p 4]

[PX 18A, p 5]

SORBO-MAT



controlled castability, machinability, density and strength



Some of the equipment in the Foundry that can reduce your costs—

3 latest type electric-induction melting furnaces.

Hydraulic-press, roll-over and match-plate squeezer molding units.

Endless-belt mold conveyor served by monorailed ladles.

Automatic shake-out devices to separate hot castings from sand and molds.

Devices that accomplish rapid removal of unwanted fins and gates.

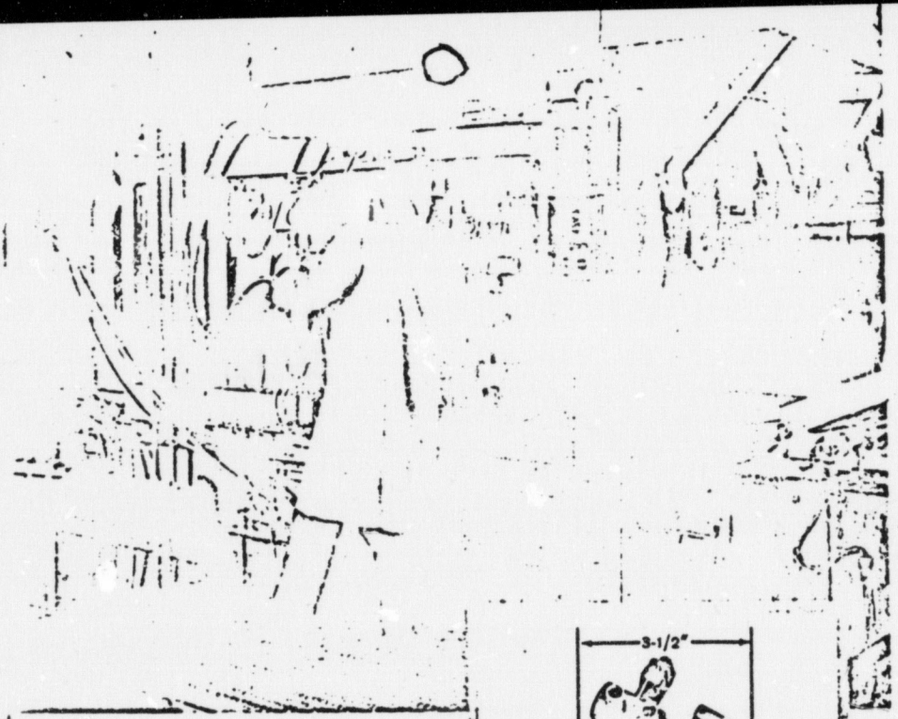
Super-Tumblast Wheelabrators to clean large quantities of castings simultaneously and efficiently.

Certainly this is one of the few foundries, in a metropolitan area, capable of pouring gray-iron constantly, day and night, into products beneficial to many varied industries.

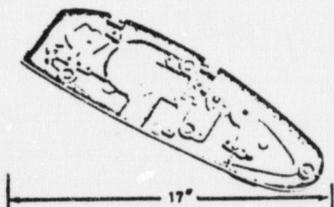
[PX 18A, p 5]

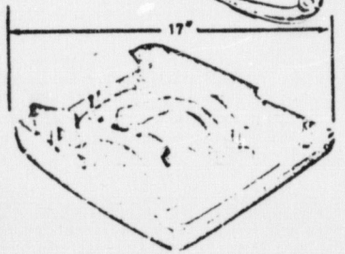
E-111

[PX 18A, p 6]



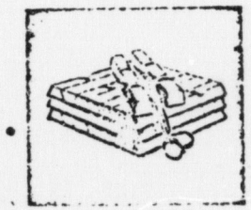






1 million pounds monthly capacity quality tested aluminum castings

Die Casting



One of the most economical methods of accurately mass-producing parts today is Die Casting—a major operation at SINGER-ELIZABETH.

Covering more than 48,000 square feet under one roof, this arsenal of production uses eight aluminum die casting machines with a locking capacity of 800 tons, eight machines with a locking strength of 600 tons, two zinc die casting machines and one 2000 lb. recovery furnace.

Armed with sixteen modern aluminum die casting machines, this department is capable of producing innumerable varieties of castings ranging in weight from one ounce to 16 pounds.

Single dies, with five or more core-pull units, as well as multiple dies, with up to eight cavities for small castings are designed, made and used within this department.

The largest dies, in use here, have a maximum casting area of 400 square inches (at 75% of fluidity) and employ an impact pressure on the molten metal of 6000 pounds per square inch. The dense, finished castings can measure 20 inches in their longest dimension.

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[PX 18A, p 6]

[PX 18A, p 7]

25 years prime quality experience behind every die casting

Each die casting machine is backed up by an electronically controlled induction furnace maintaining the molten metal at the correct temperature for casting.

Labor saving, power-driven punch and trim presses, band saws and belt sanders are in abundance, busily applying the final touches to the desired product.

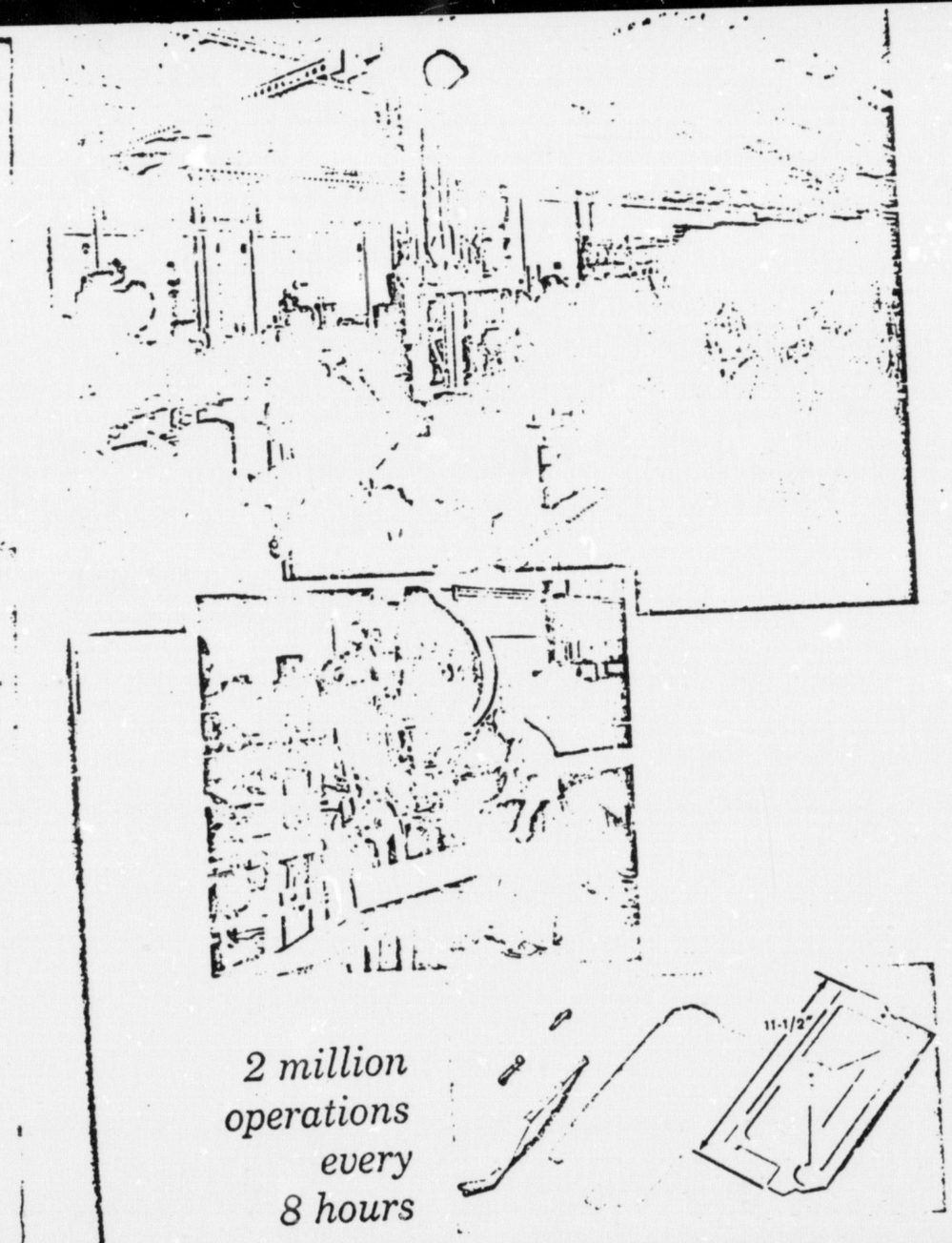
Unused gates and risers are remelted daily and after being tested for quality of content, are used again in future die casting operations.

Complete operation from ingot to shipping carton is performed quickly and efficiently under rigid quality controls. Finished castings, safely packed, are moved to the department's own shipping platform, where they are immediately transferred to railroad car, truck or customer's vehicle.

A full capacity of over one million pounds, monthly, under highly efficient methods, plus rapid, accurate mass duplication of superior castings, insures minimum costs for every customer.

[PX 18A, p 7]

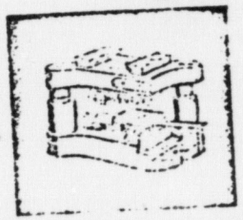
E-113



2 million operations every 8 hours

Power Press Forge

[PX 18A, p 8]



This department is physically equipped and skillfully manned to produce any item that can be pressed from five pounds or less of metal stock.

Spread out efficiently over 75,000 square feet of floor area, this most modern plant contains 200 machines, including almost every conceivable type of power press, from the powerful blanking machines to the innumerable hand presses.

Constantly alert for better, faster methods of serving the needs of customers from every type of industry, the Power Press department is a champion of SINGER integrity, quality and service.

Blanks are properly aligned and surfaces formed to within an accuracy of .002 of an inch, eliminating further expensive machining by the customer.

A supplementary tooling force within this department is quite capable of producing, repairing or modifying dies for almost any specialized or standard application.

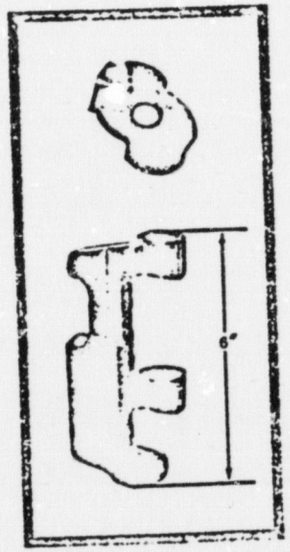
[PX 18A, p 9]

intricate designs mass produced



In the hot forging section, six hammers, striking with a force of 600 to 1500 pounds per square inch, knock out thousands of forgings each day.

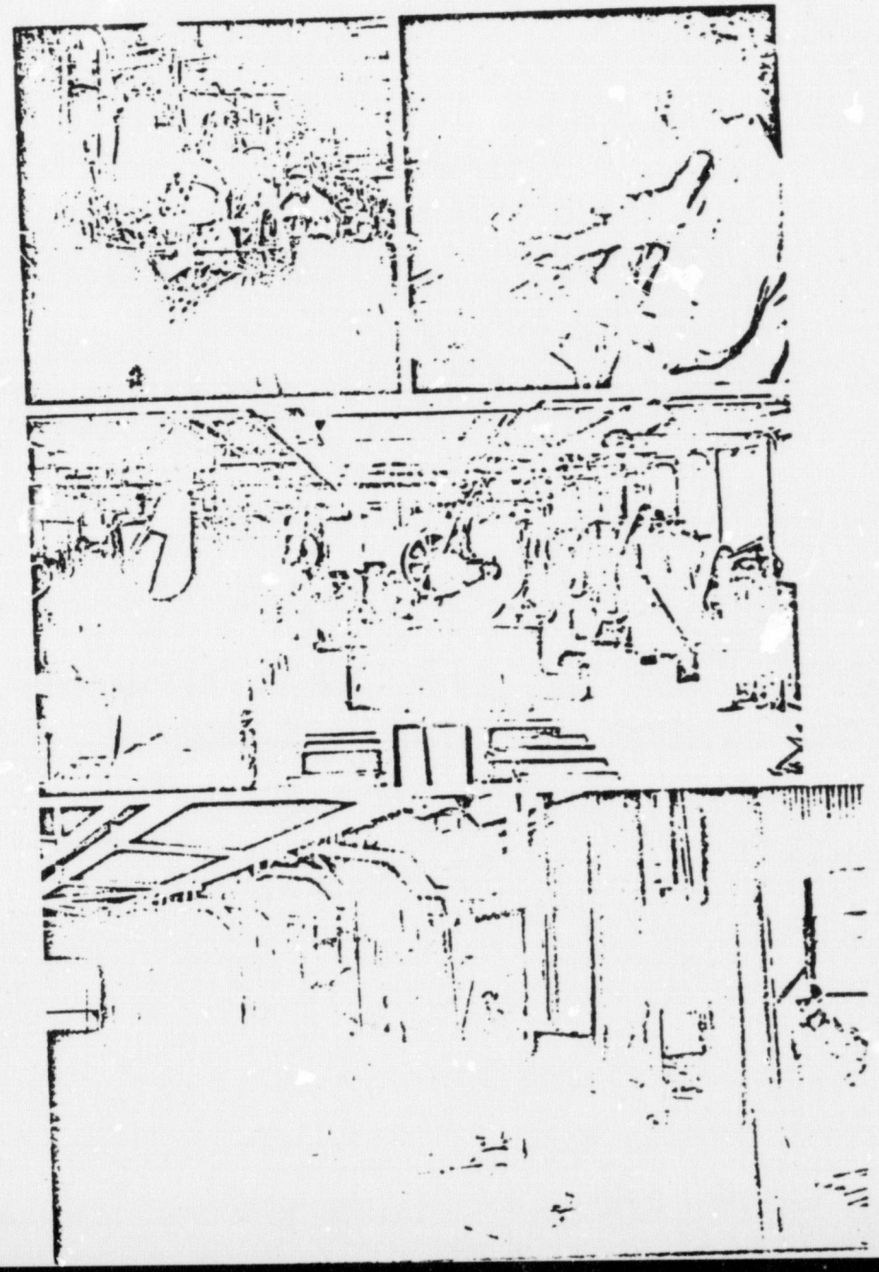
Some idea of the capacity of this facility may be grasped by a list of just a few of the machines at work here — — —



1 1500 lb. Forging Hammer
3 1000 lb. Forging Hammers
1 800 lb. Forging Hammer
1 600 lb. Forging Hammer
26 Feed-o-matic Units
1 Niagara AF 5½ (110 Ton) Press
1 Niagara U-80 (80 Ton) Press
1 Bliss 21½ M (60 Ton) Press
68 1¼ Niagara (14 Ton) Presses
1 Double Spindle Cornering Unit
3 High Speed Riveting Hammers

. . . plus many other first and second stroke heavy-duty presses, cross-center machines, one to four-spindle units, bench and floor welders, sanding, lapping, drilling, tapping machines and kick presses.

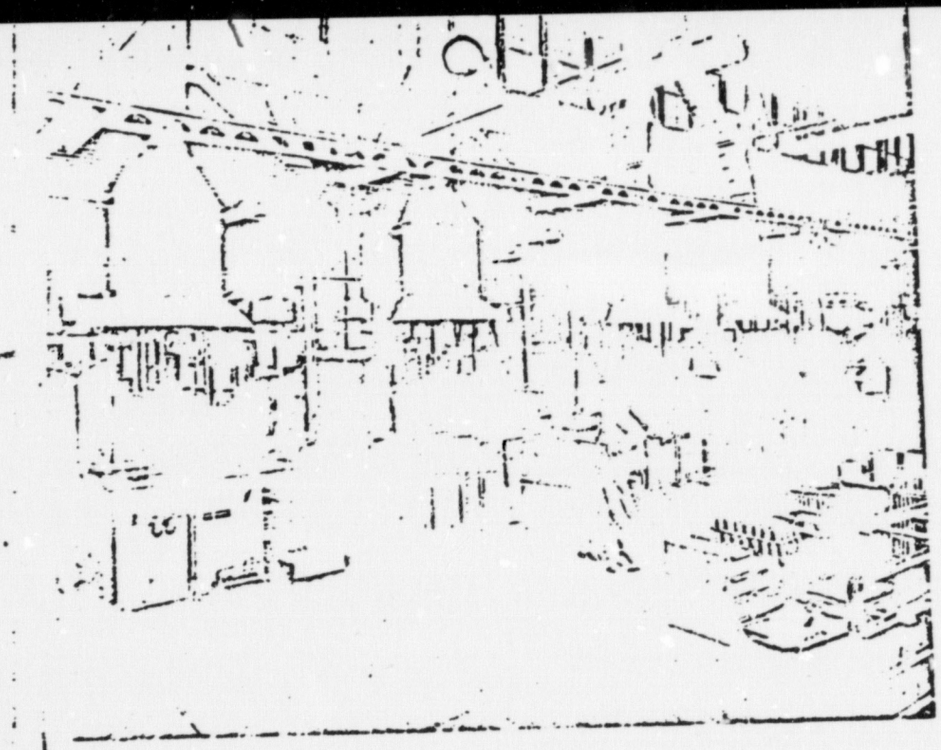
high quality forgings, capacity 5 pounds



[PX 18A, p 9]

E-115

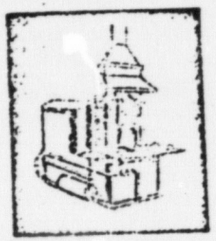
[PX 18A, p 10]



where metals are given toughness, flexibility and machinability



Hardening Annealing



Here, in the Hardening and Annealing Department, metals are given the qualities necessary to aid in machining and finishing processes and to insure the accuracy, economy and endurance of high grade tools and other products.

Coincident with the growth of SINGER-ELIZABETH, this facility, enriched in experience, has ever kept in advance of the current technical knowledge in its field.

Along with much of the latest equipment available, Hardening and Annealing Department generously employs electronic controls and conveyor lines to increase its efficiency and to decrease the cost of its products.

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[PX 18A, p 10]

[PX 18A, p 11]

invest in experience

Processes and equipment—

Hardening

Carbo-Nitriding Units

5 Leeds and Northrup Tri-Carb Furnaces (Capacity of each: 300 small parts, ½ pound or less, net load 150 pounds, or 5 large parts, 20 pounds each, net load 100 pounds)

1 Shaker Hearth Furnace (For parts as small as ¼ inch long, ¼ inch diameter to rounds, 6 inches long and ½ inch diameter)

Electronic Heating

2 Induction Units for local hardening and annealing

Automatic Temperature, Time and Gas Atmosphere Controlled Unit

1 Homo-Carb Furnace for hardening of dies for aluminum die castings (Working diameter, 24 inches. Working depth, 36 inches)

Conveyor Spring-Hardening Furnace

For 40-Carbon screws, flat and coil springs and other medium carbon steel parts, ¼" x ¼" to ½" x 8"

Annealing

Conveyor Furnaces (Belt 12 inches wide. Height clearance 6 inches.)

For cast iron (stress relief and machinability)
For steel (stress relief, machinability, ductility)

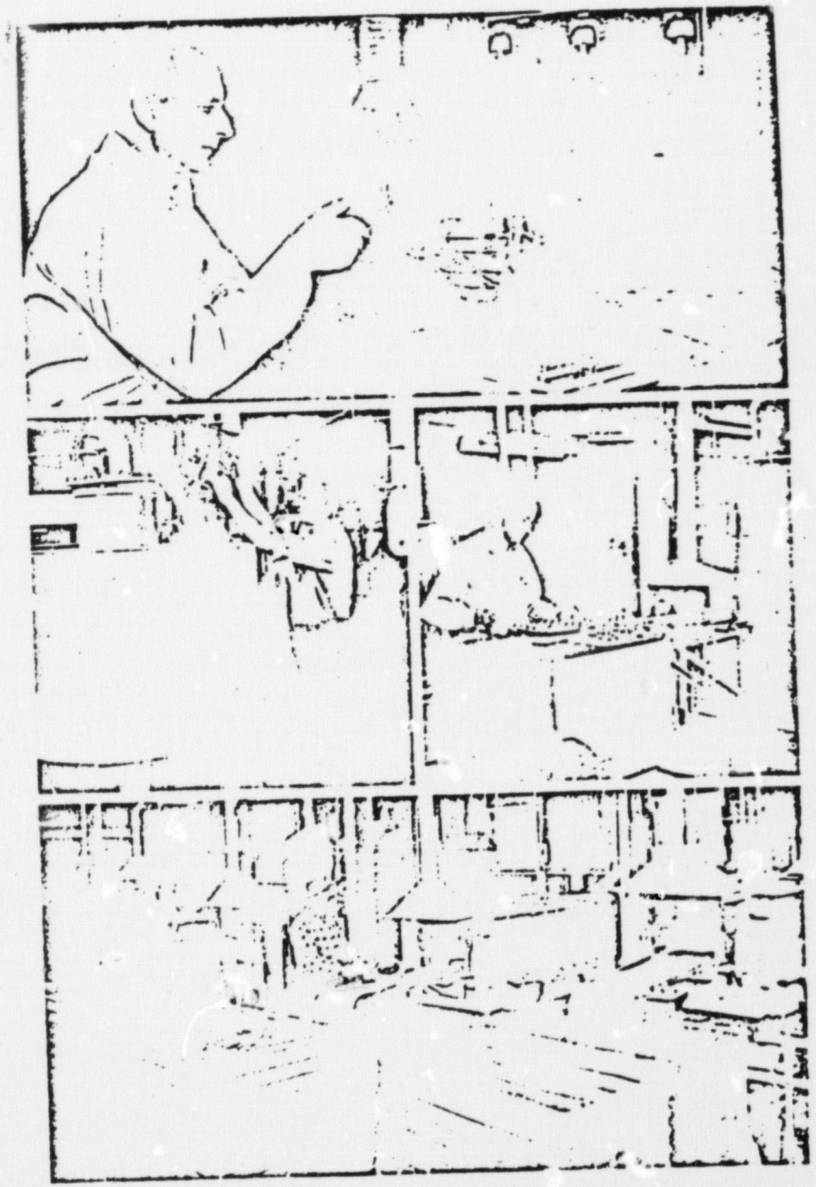
Pit-Type Furnace (Maximum 1750° F. Working diameter 25 inches. Working depth, 5 feet)

Copper Brazing and Bright Annealing Conveyor Furnaces

For low carbon steel

Tempering (Maximum working diameter, 38 inches. Working depth, 36 inches.)

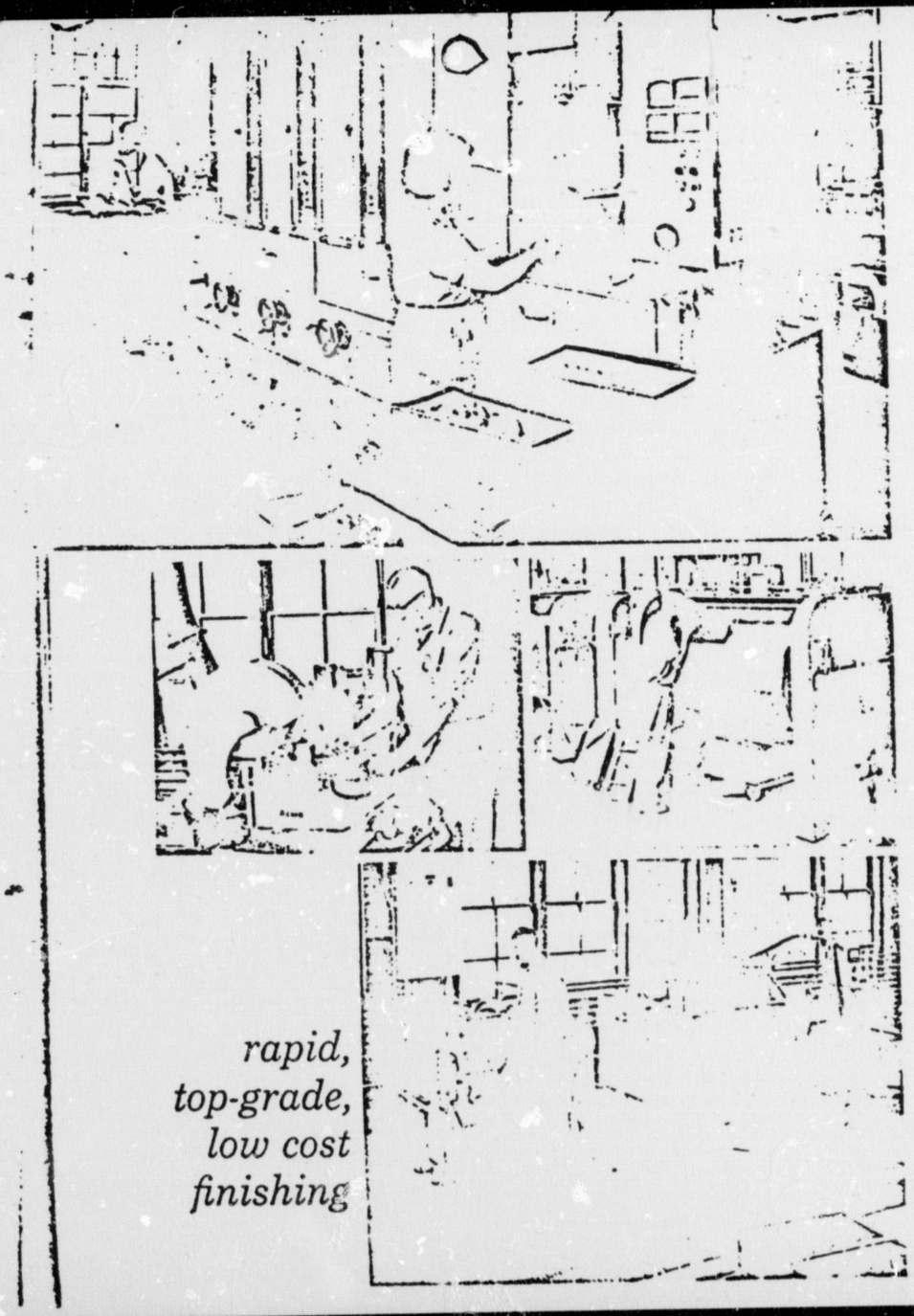
2 Pit-Type Homo-Tempering Furnaces
4 Floor Type Homo-Tempering Furnaces



[PX 18A, p 11]

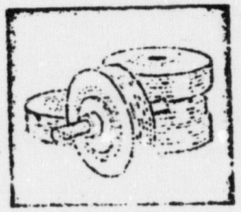
E-117

[PX 18A, p 12]



rapid, top-grade, low cost finishing

Polishing Plating



Frequently far ahead of industry, in general, in its application of Polishing and Plating processes and in its improvements in automatic machinery, this Division is extremely well organized for quality finishing of parts one foot square or smaller.

To insure the efficiency and quality of the work, an air conditioning system pumps over 300,000 cubic feet of fresh, comfort-controlled air into this area every minute.

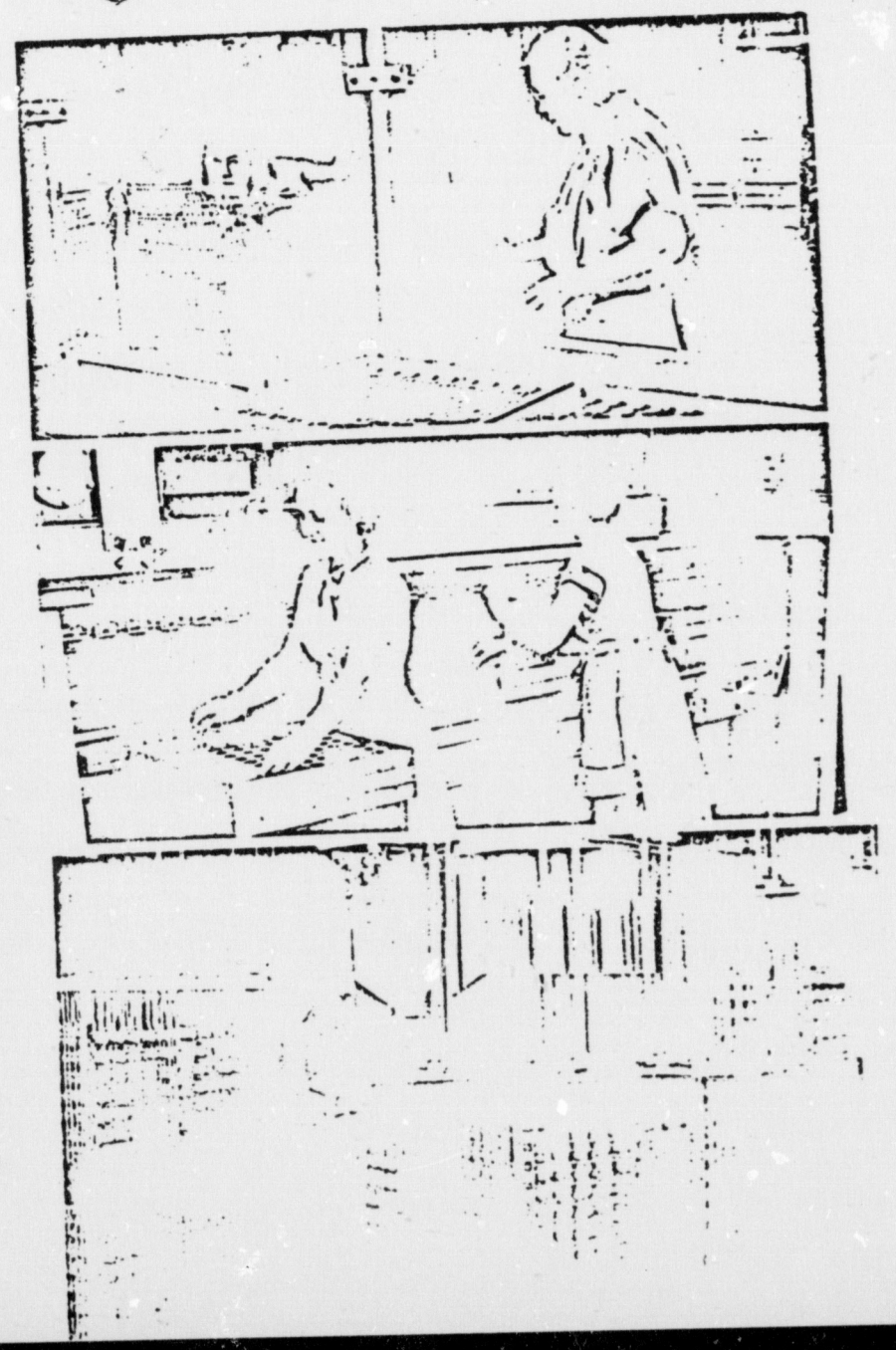
Tumbling, rotary buffing, nickel and chromium plating, anodizing and black oxidizing are expertly controlled to produce the finest finish possible on every metal product.

[PX 18A, p 12]

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[PX 18A, p 13]

over 8 million parts
finely finished weekly



Examples of SINGER-ELIZABETH conservation of time, space and production costs, coupled with improvements of finished products are the recent installations of four Lorco Vibratory Finishing Machines with capacities of one to 5.7 cubic feet and two Elliott Vibraslide Tumbling Barrels (capacity, 16 cubic feet, each). For bulk finishing of parts by means of a rotary abrasive slurry, these machines deburr, descale, grind, establish radii and polish inside and outside surfaces. Recessed, shielded and almost inaccessible areas are cleaned and polished better than by any other mechanical method known today.

Closely adhering to latest developments passed on to them by their teammates in the SINGER-ELIZABETH Metallurgical Laboratory and on the Engineering staff, the skilled personnel in Polishing and Plating are ready to serve your needs.

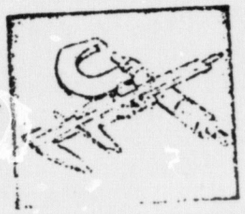
[PX 18A, p 13]

[PX 18A, p 14]



dedicated to quality, accuracy, durability

Tools Gauges



Exceptionally high standards of quality require precise instruments and tools of measurement. Control of every conceivable type of measurement begins in this department. The extreme accuracy of work performed on every piece of machinery in the plant is firmly maintained with the precision tools and gauges made here.

Manned with skilled workmen, equipped with the finest equipment and supplied with steel of the highest grade, SINGER-ELIZABETH is capable of producing within the closest limits of precision, thousands of taps, milling cutters and other small tools, while building large intricate automatic machinery and maintaining its production schedule of jigs, fixtures, gauges, tools and dies for the entire factory.

Examples of the uncanny ingenuity and fine workmanship behind every SINGER-ELIZABETH product can be shown in the Standard Precision Boring Unit on Standard Base (more than 200 of which are in use at SINGER-ELIZABETH today), the automatic indexing machines, the two-, three- or four-way machines and similar standard units for drilling and tapping, manufactured in the Tool Room.

E-120

[PX 18A, p 14]

[PX 18A, p 15]



birthplace of measurement control

Covering over 60,000 square feet of floor area the Tool and Gauge Department employs the finest equipment in its field. Its machinery includes—

29 Toolmaker's Precision Lathes—with 10 to 22 inch swing and from 30 to 78 inches between centers.
4 Speed Lathes
4 Turret Lathes
1 Heavy-Duty Vertical Lathe—with 2 to 4 foot swing and from 6 to 16 feet between centers.
21 Medium-Size Horizontal Milling Machines—with adequate Vertical Milling Attachments
13 Medium-Size Vertical Milling Machines
1 Planer-Type Milling Machine—with 12 foot table (capacity, 40 inches wide, 36 inches high)
1 Heavy-Duty Planer—with 12 foot table (capacity, 60 inches wide, 54 inches high)
1 Medium-Size Shaper
14 Cylindrical Grinders—with Internal Grinding Attachments
4 Internal Grinders—various sizes
29 Surface Grinders—various sizes up to a capacity of 2 feet wide, 3 feet high and 6 feet long
4 Thread Grinding Machines
10 Duplicating and Profiling Machines
2 Heavy-Duty Boring Machines
5 Precision Jig Boring Machines—various sizes
1 Centerless Plug Lapping Machine
4 Hole Lapping Machines
1 Planetary Disc Lapping Machine

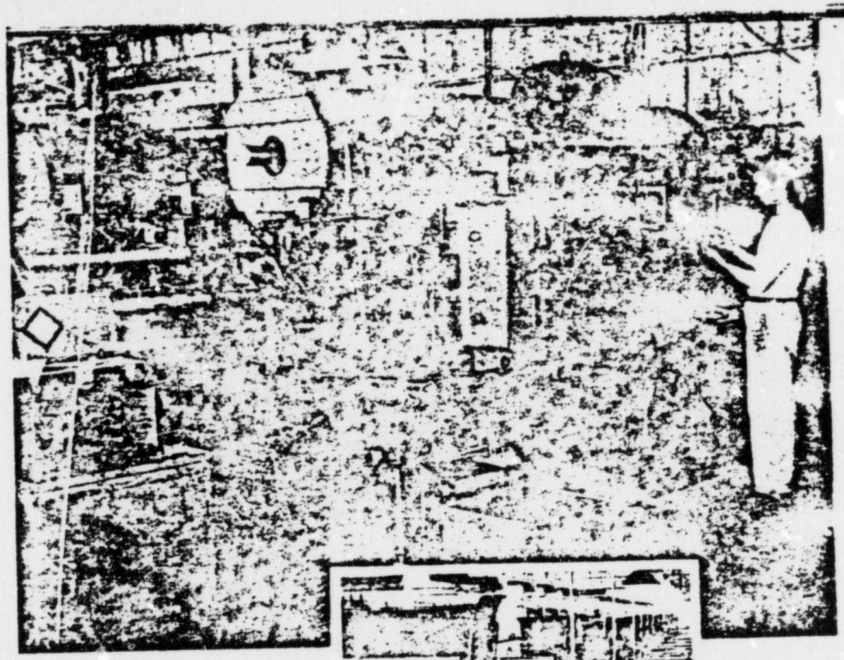
. . . plus a variety of drill presses, bench lathes, tool and cutter grinders, a precision form tool grinder and a complete complement of tool, gauge, optical and mechanical inspection equipment.

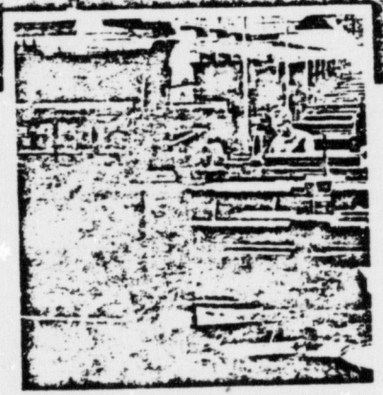


[PX 18A, p 15]

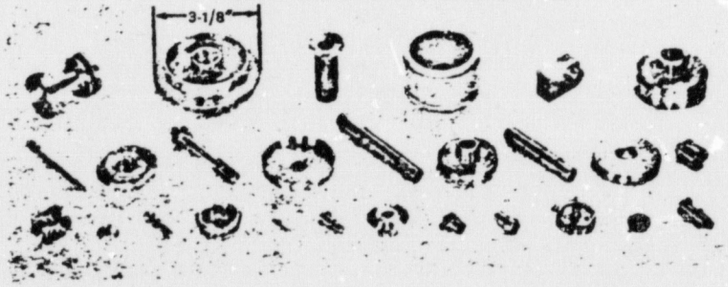
E-121

[PX 18A, p 16]

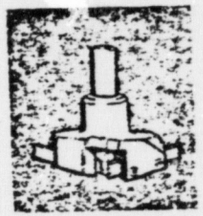




over 1 million parts daily



Screw Machines



One of the largest of its kind in the United States, the Automatic Screw Machine Department has always held a leading position throughout its history—introducing, promoting and maintaining the highest standards of mass production.

Every phase of production from contract to delivery can be performed here. Precision machines stand ready to match prescribed tolerances as fine as .001 inch. Gear cutting, milling, drilling and other secondary operations are expertly done. Skilled craftsmen with many years of experience plan the project, make the tools and guide the machines.

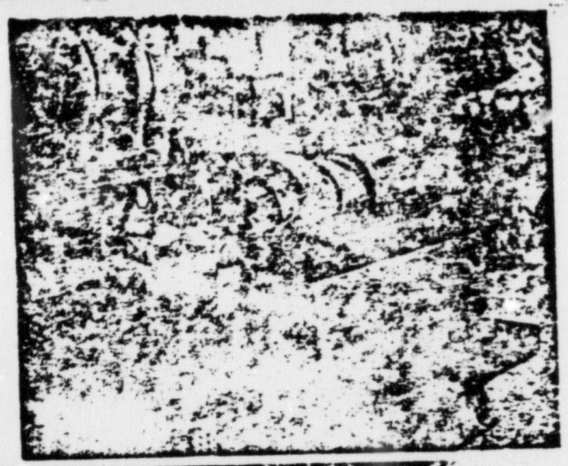
The superlative standards of SINGER-ELIZABETH are truly exemplified in the high quality of the countless products which bear this department's touch of perfection.

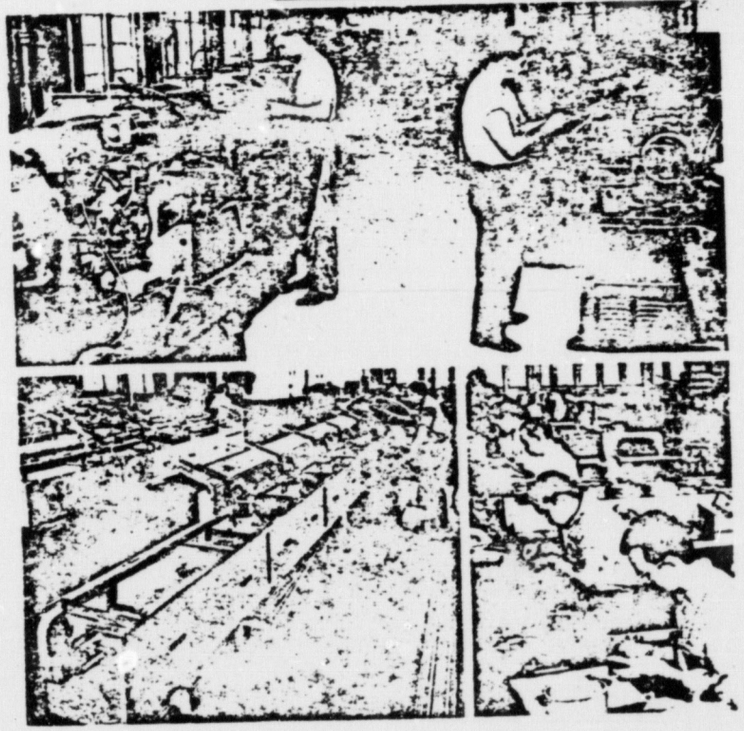
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[PX 18A, p 16]

[PX 18A, p 17]

made to your standards at minimum cost





This facility is equipped with nearly every type of automatic and semi-automatic screw machine, spread comfortably but economically over an area of 90,000 square feet.

Primary Equipment in Use at SINGER-ELIZABETH

272 Bar Stock, Single Spindle, Automatic Machines

3/8" Rd. Capacity—Brown & Sharpe	75
5/8" Rd. Capacity—Brown & Sharpe	73
1" Rd. Capacity—Brown & Sharpe	33
1 1/2" Rd. Capacity—Brown & Sharpe	76
2 3/8" Rd. Capacity—Brown & Sharpe	11
3 1/4" Rd. Capacity—Cleveland	4

46 Magazine and Hand Load, Single Spindle, Automatic Machines

3/8" Rd. Capacity—Brown & Sharpe	4
5/8" Rd. Capacity—Brown & Sharpe	7
1" Rd. Capacity—Brown & Sharpe	1
1 1/2" Rd. Capacity—Brown & Sharpe	32
2 3/8" Rd. Capacity—Brown & Sharpe	2

12 Bar Stock, Multiple Spindle, Automatic Machines

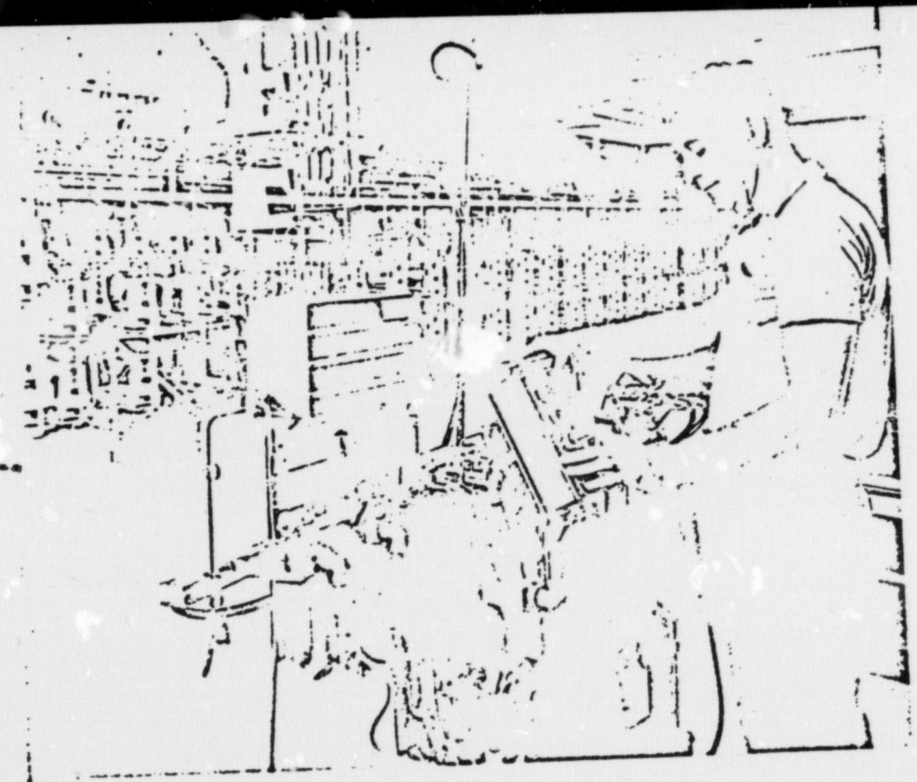
9/16" Rd. Capacity-Acme-Grindley	5
9/16" Rd. Capacity—Davenport	9
7/8" Rd. Capacity—Cone	1
1" Rd. Capacity—Greenlee	5
1 1/4" Rd. Capacity—Cone	2
1 1/2" Rd. Capacity—Cone	6
1 5/8" Rd. Capacity—Greenlee	4
2" Rd. Capacity—Greenlee	1
2" Rd. Capacity—Cone	8
2 5/8" Rd. Capacity—Cone	1

11 Hand Screw Machines—Brown & Sharpe

2 #4 Turret Lathes—Gisholt

This equipment combined with the numerous secondary machines available at SINGER-ELIZABETH, makes it possible to offer you the finest, fastest and most economical production in the automatic screw machine field.

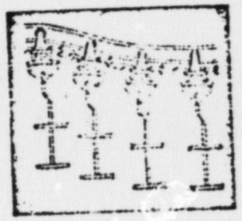
[PX 18A, p 18]





work simplification is standard procedure

Assembly



In assembly, all administration, engineering and production controls are centralized to maintain an efficient, continuous flow of parts, sub-assemblies and complete assemblies, from raw stock to shipment of products.

Highly concentrated mass production of very large orders, requiring electrostatic painting as well as assembly, is rapidly accomplished on the very modern SINGER-ELIZABETH assembly line conveyor system. However, on most of the smaller orders, the assembly of many small parts can be more easily, quickly and expertly performed in another section of the facility.

SINGER-ELIZABETH is an experienced specialist in the assembly of small parts and fulfillment of small emergency orders.

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[PX 18A, p 18]

[PX 18A, p 19]

advanced modern equipment spells sure quality at minimum cost

Many of the automatic fabricating and assembling machines used here were designed and built within this plant. Their fine workmanship and delicate accuracy assures highest quality products at lowest possible cost.

Since standardization of parts has kept SINGER strong in competition, this principle is carried out to the utmost degree in Assembly; insuring speed and economy.

One of their most complex parts, with tolerances as fine as .0001 inch, requires 90 operations. Although all tolerances are met, many thousands of parts are made and assembled each day.

Inspection has taken on the modern role of advisor and production aid rather than that of ruthless rejector. Standards of quality are integrated with design of assembly before the initial production of any one product. Methods of maintaining high quality may be visually evident in the use of electric comparators, optical comparators, air gauges and other types of direct reading gauges at frequent points along the assembly lines.



[PX 18A, p 19]



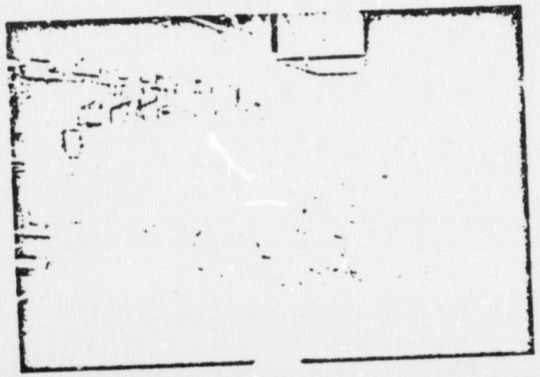
Electrostatic Painting



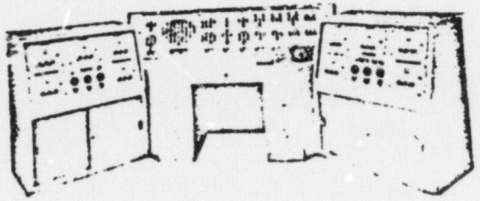
Modern custom painting, economical hand and semi-automatic spraying, as well as electrostatic painting are expertly accomplished in this area.

The Electrostatic Painting process consists of charging the paint solution with 100,000 volts of electricity and then atomizing it with rapidly revolving discs. The items to be painted are grounded to the conveyor, to attract just the right coating of paint before passing on to the flash-off chamber. There the combustible solvents are evaporated. Baking ovens fix the coating. Each coating is allowed to cool before the process is repeated. As many coats as are required may be applied in this manner before the final baking is accomplished. All of these operations are executed in a pressurized, dust-free area.

designed to increase sales potential



[PX 18A, p 21]



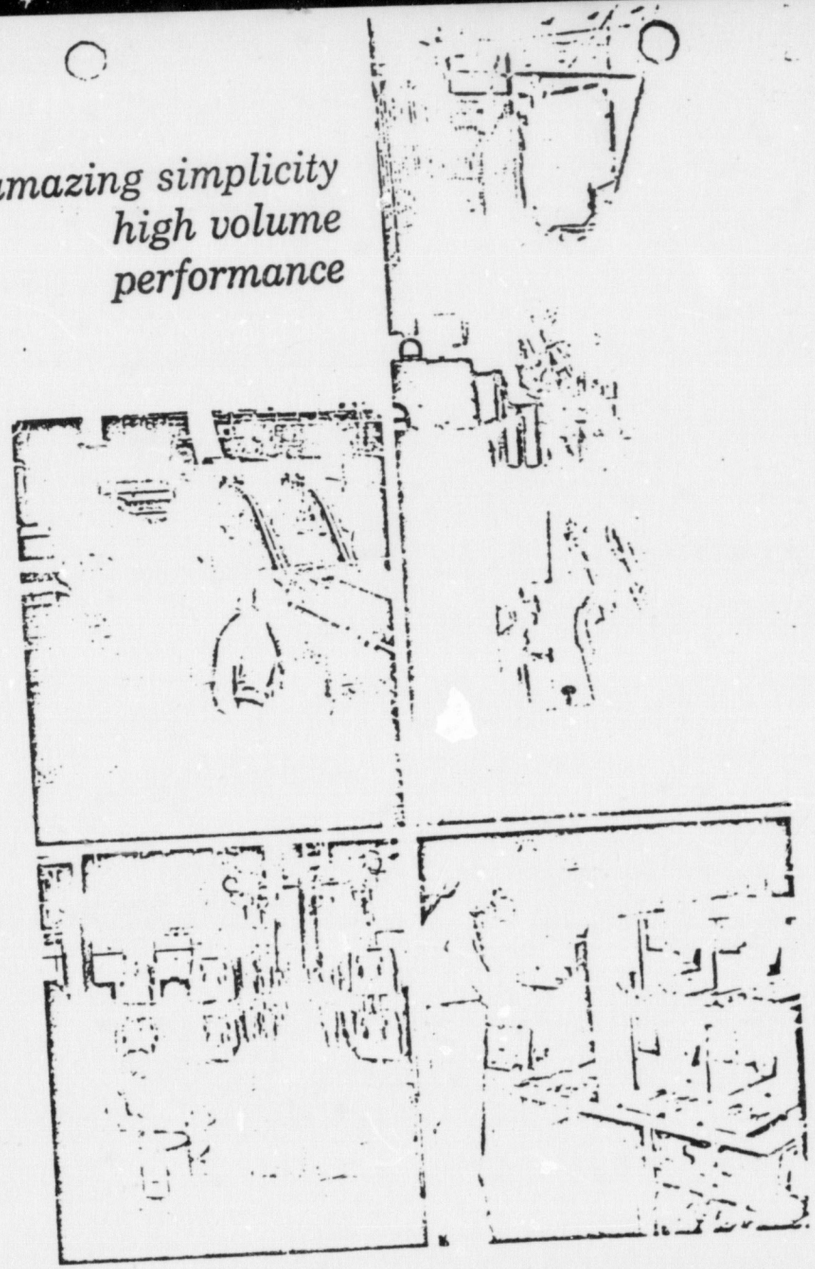
The conveyor system performs an important task in Electrostatic Painting—taking parts and assemblies through degreasing and joining operations, through cleaning and chemical preparations for painting and finally through the electrostatic painting cells, the flash-off chambers and the baking ovens.

It is fitting that the final results of careful planning, skilled workmanship and high quality controls should be decoratively finished by the most modern means of painting in use today.

Smaller orders are expertly and more economically accomplished in the hand spraying and semi-automatic spraying areas.

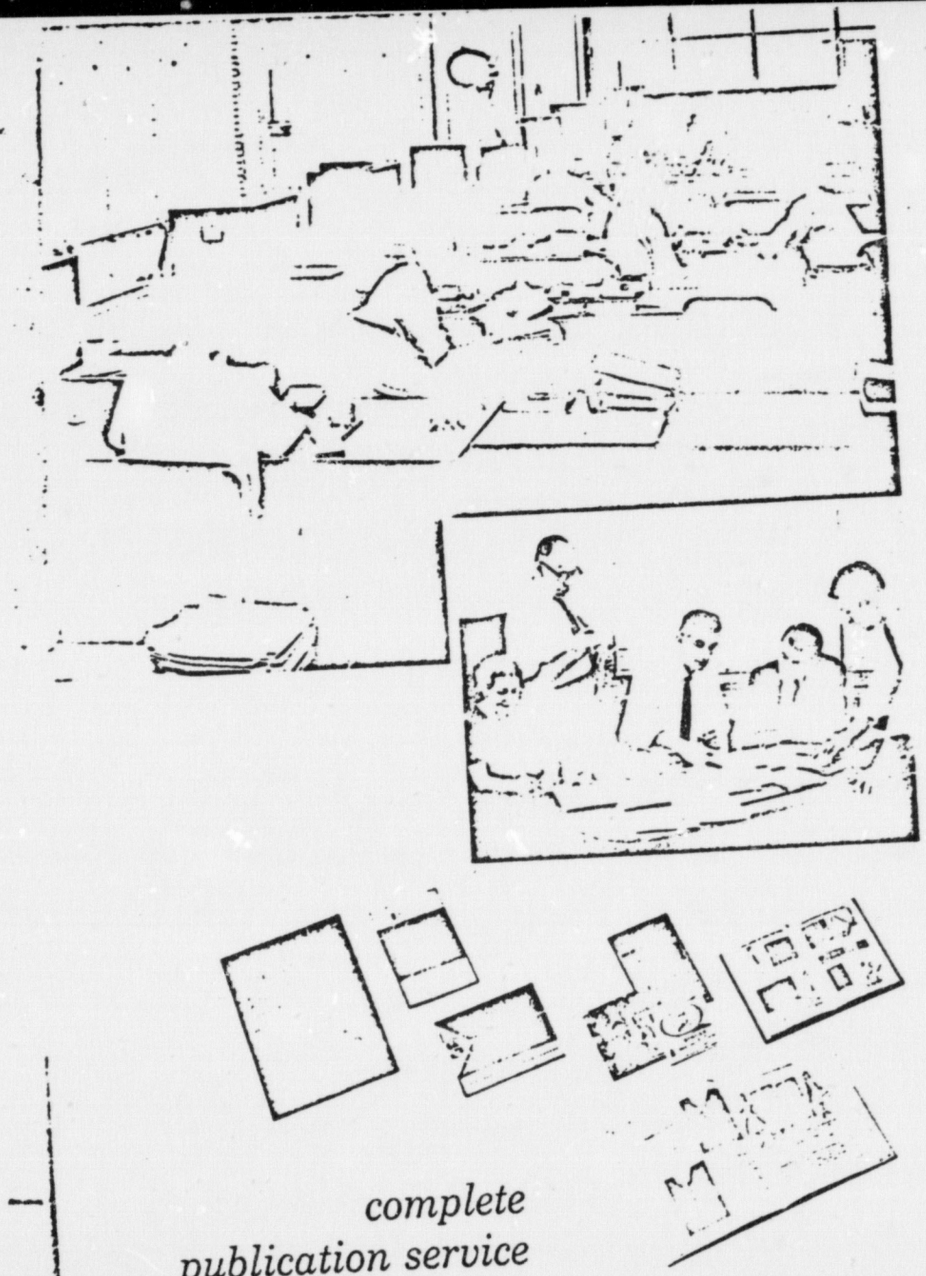
Ornamentation, applied by means of silk screening, is baked in infra-red ovens.

amazing simplicity
high volume
performance



E-127

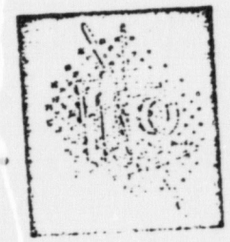
[PX 18A, p 21]



complete
publication service

[PX 18A, p 22]

Publications



The Technical Publications and Graphic Arts Service at SINGER-ELIZABETH is capable of satisfying every editorial desire of the modern manufacturer.

Fulfilling the needs of the World-Wide SINGER organization throughout the years, its publications have been printed in over 50 different languages. So wide and intense has been the distribution of its literature and artwork, along with the Company's products, that the trademark SINGER® has long become a household word in every corner of the globe.

As with all other SINGER-ELIZABETH MANUFACTURING SERVICES, quality and customer satisfaction have always been the main objectives.

Whether you need a tiny label, a conventional photo, a fancy piece of artwork or a brochure such as this, SINGER-ELIZABETH is confident of its ability to fulfill your need quickly, at minimum expense. Call or write for further information that may increase your sales potential.

[PX 18A, p 22]

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This Department's services include—

Technical, institutional and promotional literature from planning to printing—layout, writing, editing, illustrating and film processing.

Technical and commercial art from creative layout to final artwork—airbrush rendering, photo retouching, line drawings, phantom and perspective views, graphics and mechanical paste-ups.

Photography—instructional, merchandising, personnel, public relations and architectural. Print capacity, 6000 per week.

Photo Lithography—line and halftone combination, multiple Benday tints, duotone negatives, color separation from line copy, letterpress negatives and film correction including spotting, stripping and multiple-color ruling. Camera capacity (ATF Automatic Process) 1000 20" x 24" line films per week.

Among the more interesting achievements of this department—

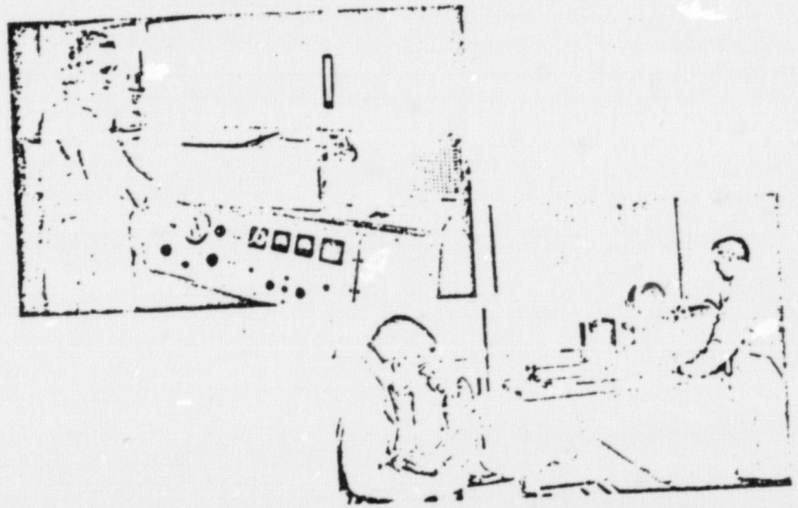
The textless instruction book, which eliminates the necessity of language translation, permitting communication with all peoples no matter how great the language barrier. These books have met with instant, unqualified success from their inception.

The condensation of multiple-page operator's instructions to a simple card form by means of visual aid techniques.

[PX 18A, p 23]



Metallurgical Laboratory



Supporting Facilities

Directly affiliated with every department presented in this brochure, the Metallurgical Laboratory establishes and maintains quality of materials used in SINGER-ELIZABETH production processes from pig-iron to paint. Its primary function is to investigate, develop and test new manufacturing processes and to determine their possible application to production needs.

Each shipment of raw materials to SINGER-ELIZABETH is carefully sampled, examined and tested. All materials are spot-checked during processing periodically by the Laboratory staff to maintain established standards of quality.

The Metallurgical Laboratory is expertly manned and completely equipped for every requirement of chemical, physical and visual analysis within its field. Microscopic examinations are accomplished through the use of their micro-metallograph and photomicrograph units.

Paint and metal finishes and anti-corrosive coatings are methodically evaluated for the particular sets of conditions of weather and wear to be met by each product.

[PX 18A, p 24]

[PX 18A, p 25]

Engineering • Designing

Since Isaac Merritt Singer created a machine in 1850 that revolutionized the entire sewing industry, men of vision and leadership have occupied a preeminent position in the Singer Company.

A history of prompt, practical solutions to varied complex problems has earned SINGER-ELIZABETH Engineers a reputation for sound ability and knowhow. Many of their past and present experiences may parallel your own requirements.

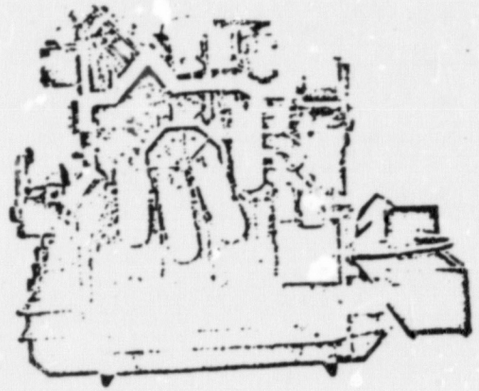
The present staff of 42 Engineers, plus 12 Designers and Draftsmen, applies a high degree of engineering ability to the design and contruction of special automatic machinery. Such machines as the SINGER-ELIZABETH Standard Units perform any desired combination of operations including drilling, reaming, tapping, boring, facing, turning and milling.

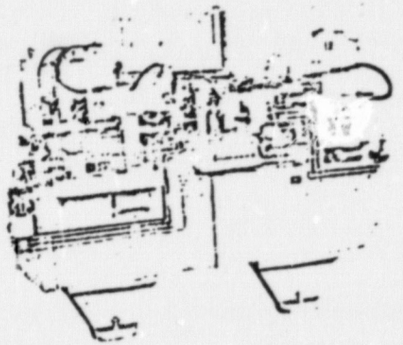
SINGER-ELIZABETH Engineers actively cooperate with manufacturers of standard types of production machines, designing and constructing special adaptations and units to obtain more economic and comprehensive automatic machining of high-production SINGER-ELIZABETH products.

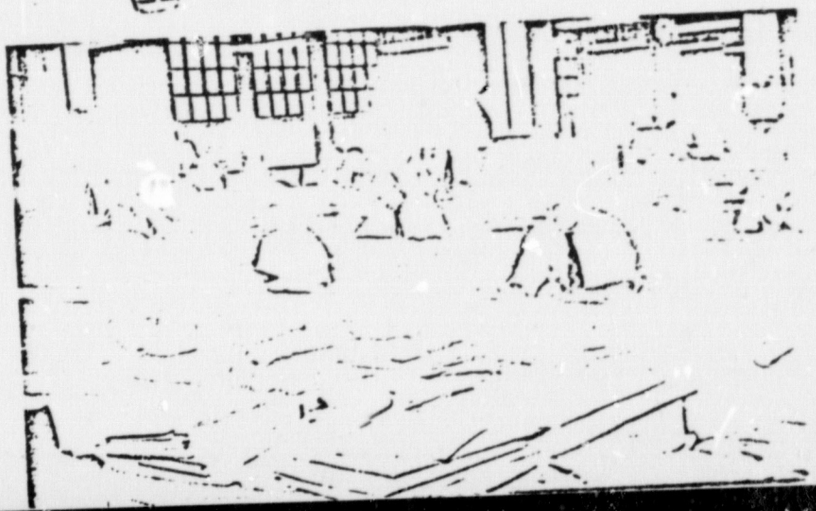
One example of this joint exercise of SINGER-ELIZABETH Engineers and their machinery suppliers was the cooperative designing of the 1½ mile Continuous Traveling Conveyor and Automatic Transfer System in the SINGER-ELIZABETH Sewing Machine Assembly Plant. From die castings of arms and beds to shipping cartons this Assembly Line is the last word in modern efficiency, speed and economy.

One section of the Engineering staff studies new devices and parts to enable them to define the material, method, procedure and the equipment to be used in manufacturing processes. Their alertness to latest developments in the field of product manufacture and their background of long experience assure dependable modern manufacture. Behind this group stands a history of manufacturing engineering for the federal government and for other manufacturers.

The Engineers are ready and able to render prompt, expert assistance to all stages of product development and of equipment maintenance, modification and repairs.







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[PX 18A, p 25]

[PX 18A, p 26]

Quality Control

Quality is not accidental. It must be planned and built into a product.

Quality depends on people. The people in this supporting facility, ever on the alert, demand and achieve the quality synonymous with SINGER.

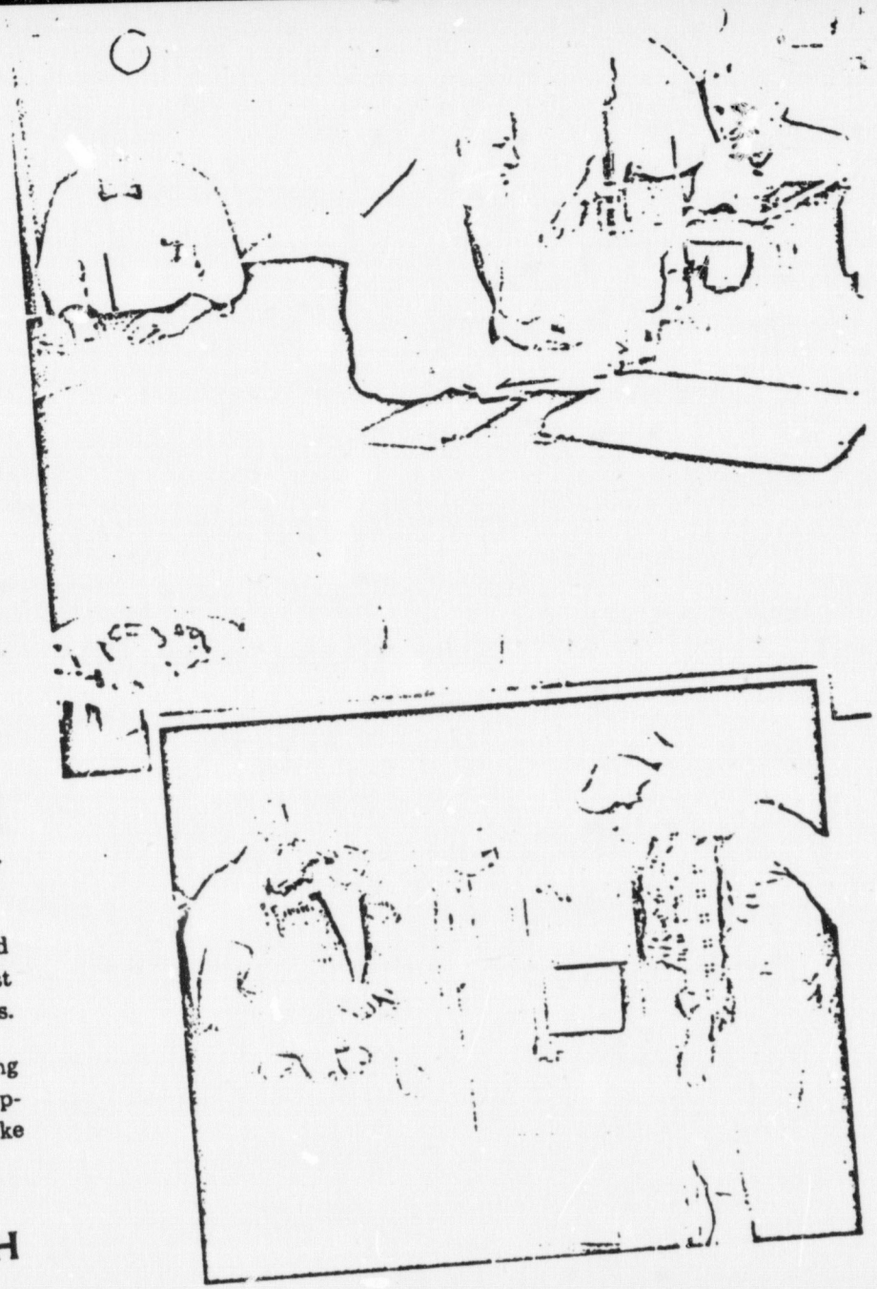
All inspection controls are based upon current statistical data and techniques. A Quality Control Manual has been issued by this department to formalize procedures, methods and instructions. It is used as a guide in performing quality control functions. Military Standard #105-C, Sampling Procedures and Tables for Inspection by Attributes, is also used.

A unique feature of the Quality Control Department is its ability to detect a defect early in production—perhaps as early as the engineering and laboratory phases of planning. Such an accomplishment can eliminate a staggering amount of scrap and re-work.

This team of workers, set-up men, supervisors, engineers and inspectors acts unfailingly to effect quality improvements and cost reduction, providing end products that satisfy customer requirements.

Along with the Metallurgical Laboratory and the Engineering staff, the SINGER-ELIZABETH Quality Control Department upholds the sound ideals of top-quality and low costs which make SINGER products praised and desired the world over.

grow with **SINGER-ELIZABETH**



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[PX 18A, p 28]

SINGER-ELIZABETH DIVISION

SINGER

THE SINGER MANUFACTURING COMPANY
321 First Street, Elizabeth, New Jersey • EL 2-5200

Furniture Stores Department

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MOTOR VEHICLE RESEARCH

Report No. 13

Standard Passenger Car Brake System Modification Produces New Safety Brake System

A Non-Profit Organization in the Interest of Automotive Safety and Research

Vol. 1 No. 13 | Published at Lee, N. H. | LEE, N. H. 1964

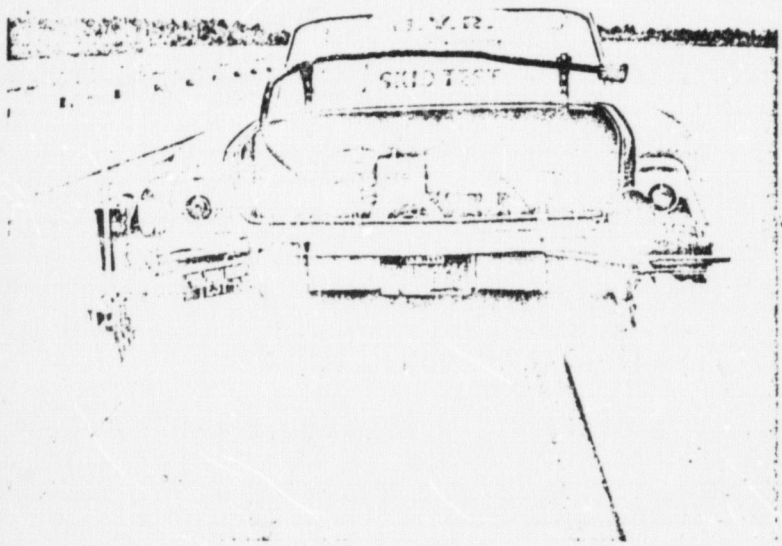
© 1964 by Andrew J. White

$1.00 per copy 12 issues $8.00

PLAINTIFF'S EXHIBIT 19

MOTOR VEHICLE RESEARCH OF NEW HAMPSHIRE Announces Spectacular Results From Tests Of New Passenger Car Braking System

System Features Anti-skid Control, Provides Steering Control and Shorter Stopping Distances



View of 1963 test vehicle. Generator to power time-motion high speed moving picture camera is located in trunk- exhaust pipe and muffler are seen positioned above generator — camera, timer, detonator, legend board, flash signal unit and reference standard are on outrigger frames.

Text of address by director of Motor Vehicle Research of New Hampshire. Safety Meeting, Heliport Building. New York Worlds Fair Grounds. November 12, 1963

Ever since man discovered the wheel some 6 thousand years ago, the progress of civilization has advanced through the transportation media. This progress has accelerated rapidly during the last 5 decades with the advent of the automobile.

For centuries man's speed spectrum had been confined to the area of 10 miles per hour then machines were discovered to replace the horse.

Even when the horse was hitched to a wagon to form a transport system for goods, the problem of stopping the wagon became clear to the driver every time the system was used. The problem was solved, to some degree, by utilizing the system of a wheel brake consisting of a shoe being forced against the wheel rim by means of a lever arm.

When the automobile was developed, this system was utilized and constantly modified to keep abreast of the advancement of vehicle design. The shoe against the wheel principle is still used in automobiles and trucks whether the brake design is a shoe-and-drum type or a disc.

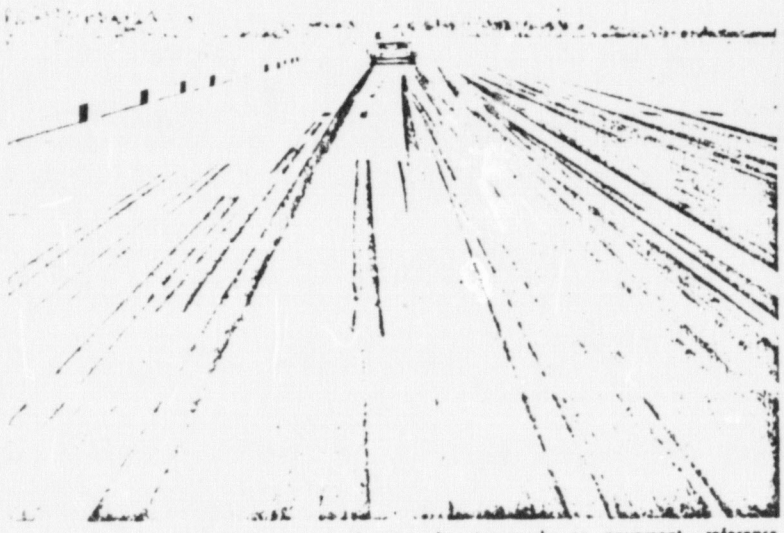
The brake actuating principle using hydraulics, air, vacuum, etc., is still the same.

This consists of forcing a shoe or pad against a drum or disc to create friction which in turn is a product of the energy of motion of the vehicle. The principle of energy conversion into heat is what slows down the vehicle when the brakes are applied.

This system of stopping vehicles has been somewhat satisfactory and is controlled by several laws of physics and mechanics which up to this time, are unalterable.

In order to discuss some of these laws in simple terms, it is necessary to cite some examples. These will be for illustrative purposes only.

Let us consider a large truck with 50,000 lbs. of payload traveling at a speed of 60 mph and a boy on a bicycle pedaling at the same speed beside the truck. Both have brakes on all wheels and tires that produce equal coefficients of friction. They both apply brakes locking all wheels and come to a stop by sliding the tires on the pavement. Which will stop faster? Both will stop at the same time as we have assigned equal coefficients of friction to the tires on both vehicles.



...e section of test area showing extent of testing by tire marks on pavement — reference ...located on left side for camera on test car to photograph.

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While this example is of a simple nature, it serves to outline the theory.

In practice however, there are a number of factors that control the stopping distance, some of which are as follows:

(1) Solid interference or keying of the tire tread to the road irregularities to create a mechanical action.

(2) The resistance to sheer of the tread stock and the hysteresis of the polymer used in that tread stock.

(3) The plastic flow and lubricating qualities of the polymer at various temperatures.

(4) The wheel load on the individual wheels controlled by load distribution and weight shift during deceleration.

While these are not the total factors controlling the stopability of the vehicles, they are the primary factors involved.

A discussion of the basic factors, physics and mechanics included, would require several years of intense study.

The stopability of a vehicle is controlled by the coefficient of friction between the tires and the road. A vehicle with the finest brakes possible to produce are next to useless in stopping a vehicle on certain types of wet ice and other surfaces.

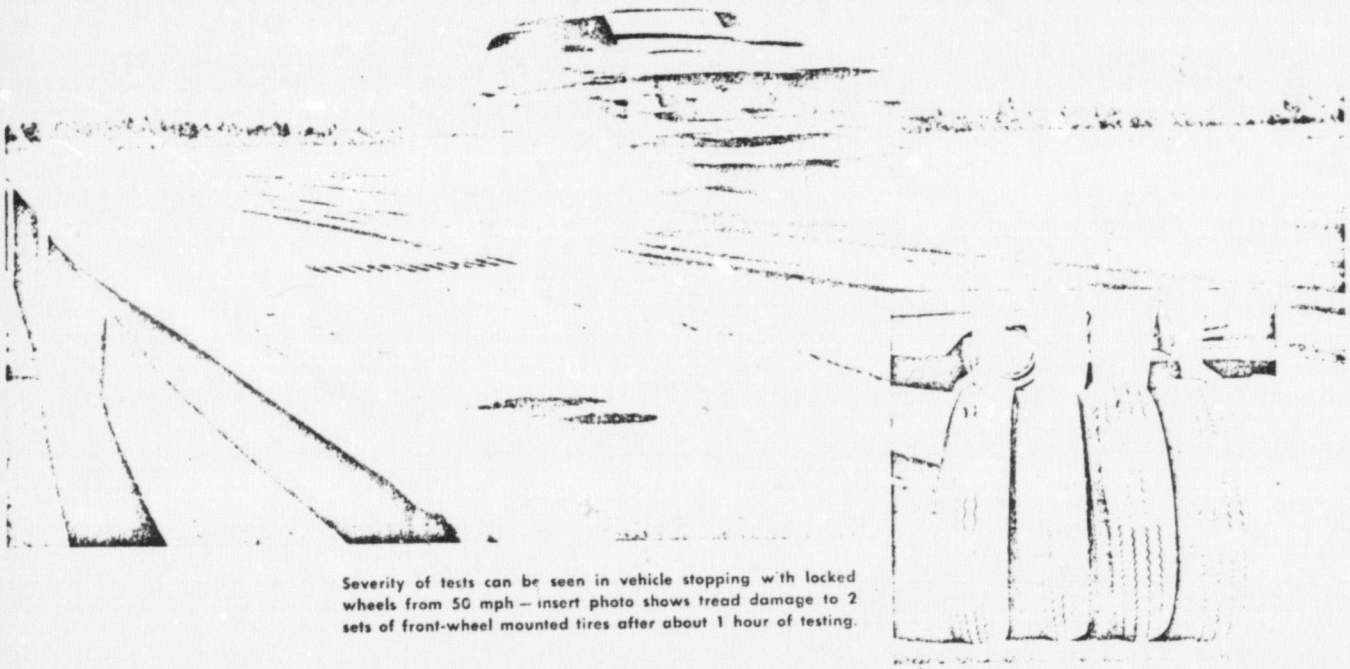
The same vehicle on a dry road can stop in a relatively short distance. In this instance, we have another factor that influences the stopability. This is wheel rotation or what is commonly termed, impending wheel skidding.

It is a recognized fact that a wheel that is rotating on the road, while being retarded by brake action just short of lock-up, can stop faster than a wheel that is locked up and sliding on the roadway.

The principles involved are widely publicized in the popular

(Continued on Page 2)

[PX 19, p 1]



Severity of tests can be seen in vehicle stopping with locked wheels from 50 mph — insert photo shows tread damage to 2 sets of front-wheel mounted tires after about 1 hour of testing.

(Continued from Page 1)

theory that pumping the brakes on a wet or icy surface is better than fully applying the brakes and locking up the wheels.

However, we are now introducing a control which is variable and too slow under emergency conditions. This control is human.

How many people are capable of pumping brakes sufficiently fast to stop a vehicle in a matter of seconds, especially when they have crash awareness tension? The answer is, very few.

Many of you here today have traveled in our modern jet aircraft. How much thought have you given to the braking systems of such vehicles?

They are equipped with braking systems that control wheel rotation during the landing roll. These systems are mandatory for safe landings under a number of conditions. They are costly, intricate and reliable, but necessary.

We are fast approaching the need for such systems for our highway vehicles. As speeds increase, the need for better control

Motor Vehicle Research of N. H. has been evaluating the Perma Anti-Skid device for a period of almost 5 years.

Our first exposure to the early prototypes of this system caused us to advise the inventors to instigate a long range research and development program in an attempt to eliminate a number of problems that were present in the system.

These early designs were crude and inadequate in satisfying the design requirements for a reliable anti-skid system. In fact, the first designs submitted and tested were far from the area of reliability and performance and as such, were not suitable for consideration.

Several other designs were submitted during recent years, all of which were rejected after field and laboratory tests.

Several months ago, a new design was submitted for evaluation and a rigid test program has been completed, in part, at this date. The design has withstood rigid laboratory and field testing.



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Man has changed little in centuries and will change little in countless centuries to come. The progression of the machine has been unending while man's progression has been slow. We must then tailor our machines to cope with man's inabilities.

In the area of stopping automobiles, we have arrived at a point where an automatic braking control is necessary. As average speeds increase, the need will become more pronounced. The legal speed sign of the future could very well read "Speed Limit 100 MPH" or even higher. Posted limits on some highways are now 80 MPH.

A few years ago, a 50 MPH sign was rare. Fifty years ago, 20 MPH was normal and before that, 10 MPH was mandatory by law.

The problems of highway safety are closely allied with vehicle braking. If this were not so, we would not have demolished vehicles at the termination of skid marks.

In this vein, the ability of an operator to steer his vehicle while braking is of vital importance in avoiding impact to another vehicle and at the same time reducing the vehicle speed at a maximum rate.

Our present braking systems do not satisfy these requirements. Thus, the need of a controlled system is mandatory.

The industry here and abroad have recognized this need and the problem has been burning for a number of years in most experimental laboratories.

Such a system is possible as seen in our aircraft braking systems but are at costs to apply to the average passenger car. Others are being developed and tested by every car and brake manufacturer throughout the world.

I have worked on the aircraft and automobile type as far back as 1951 with Prof. Jankowski of Poland, whose design included pendulum mechanism to control deceleration rates. This principle in part has been used in the over-ride fly wheel units now installed in the aircraft systems.

The early attempts were crude and unreliable, usually controlled by a centrifugal mechanism of the vehicle. It was quite evident that additional mechanical and hydraulic principles were needed to produce an acceptable system.

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In one sample, one member of the drive-train system of the anti-skid unit was sintered brass for composition and this gear functioned with a hardened steel gear. This gear combination was in constant mesh when the anti-skid unit was installed in a vehicle operating on the highway.

Laboratory tests at a constant speed of 65 miles per hour disclosed failures of the brass gear at approximately 16,000 miles. This gear was then replaced with a DuPont Delrin gear and run against a steel gear for 140,000 miles without failure.

This test is continuing to determine the life span of this combination of a gear train. Operation without failure for 140,000 miles is a satisfactory life-span for such a gear train. Inspection of these gears at this time indicates a possible life span of over 200,000 miles.

Additional tests in temperatures of 35° below zero and 250° above zero are being conducted. Also, a cycling test of the total system accelerating from 0 to 90 miles per hour and then down to zero in 3 seconds is past the 30,000 cycle mark and continuing. This test is equal to 30,000 emergency stops from 90 miles per hour.

All these laboratory tests are run under rigid specifications and the complete results will be reported in book form in the near future. Such tests also serve to prove the design of the system.

We have also completed a series of tests using a standard vehicle stopping on a crushed rock and asphalt roadway. Speeds were 20-30-40-50 and 60 miles per hour.

We have carefully recorded the stopping distances of a vehicle with and without the device. The results show a marked improvement in stopping distances of a vehicle equipped with the modification of the brake system by means of the device.

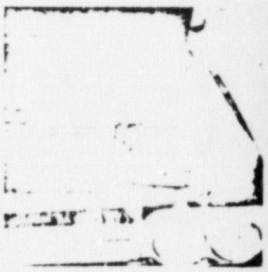
The test procedures include high speed motion picture cameras and timers recording in increments of 1000th of a second to eliminate any errors.

What has occurred is what tire engineers have sought for many years. It is possible to stop a vehicle in a shorter distance with the controlled braking system than with a standard system under emergency panic stop conditions.

I take direct exception to referring to this system solely as an

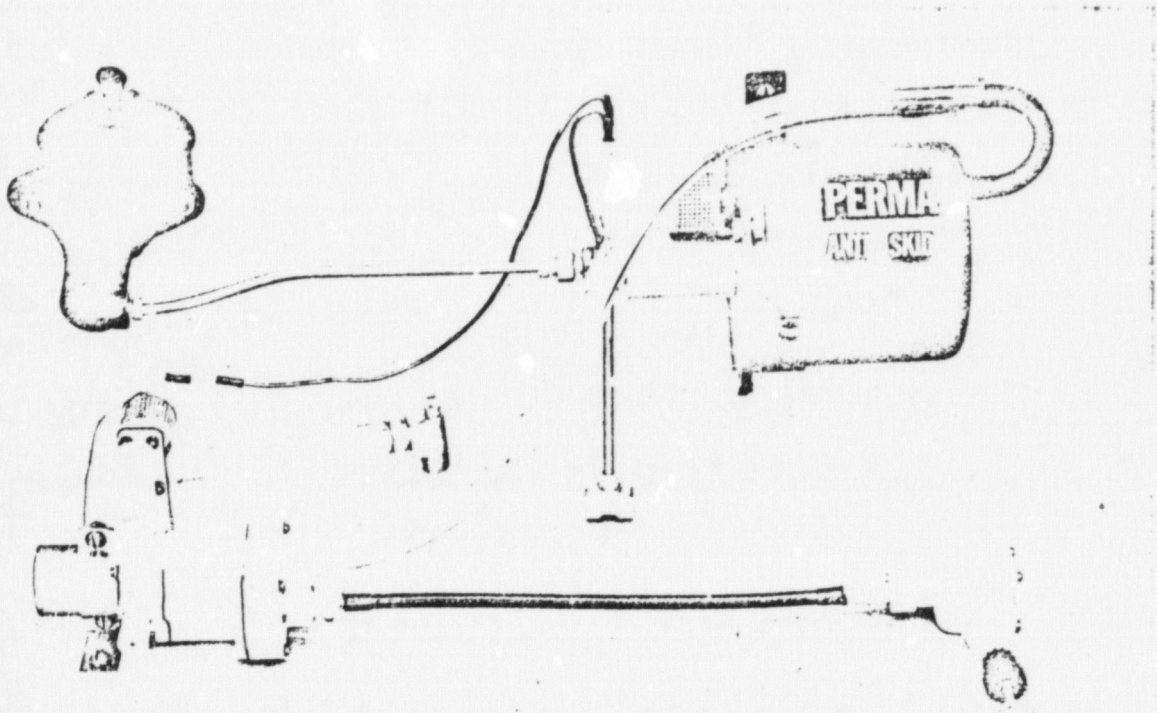
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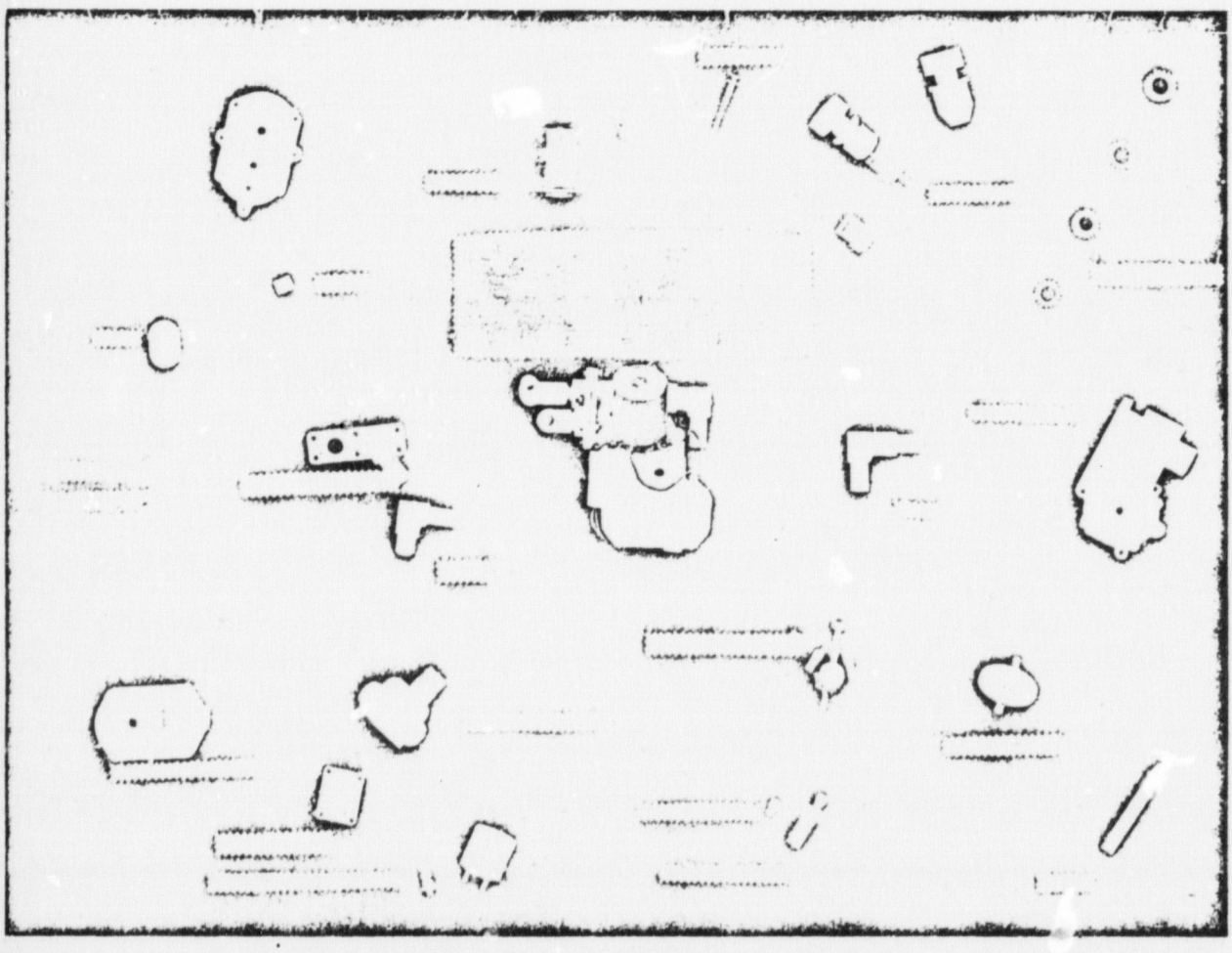




Photos above show typical under hood installations

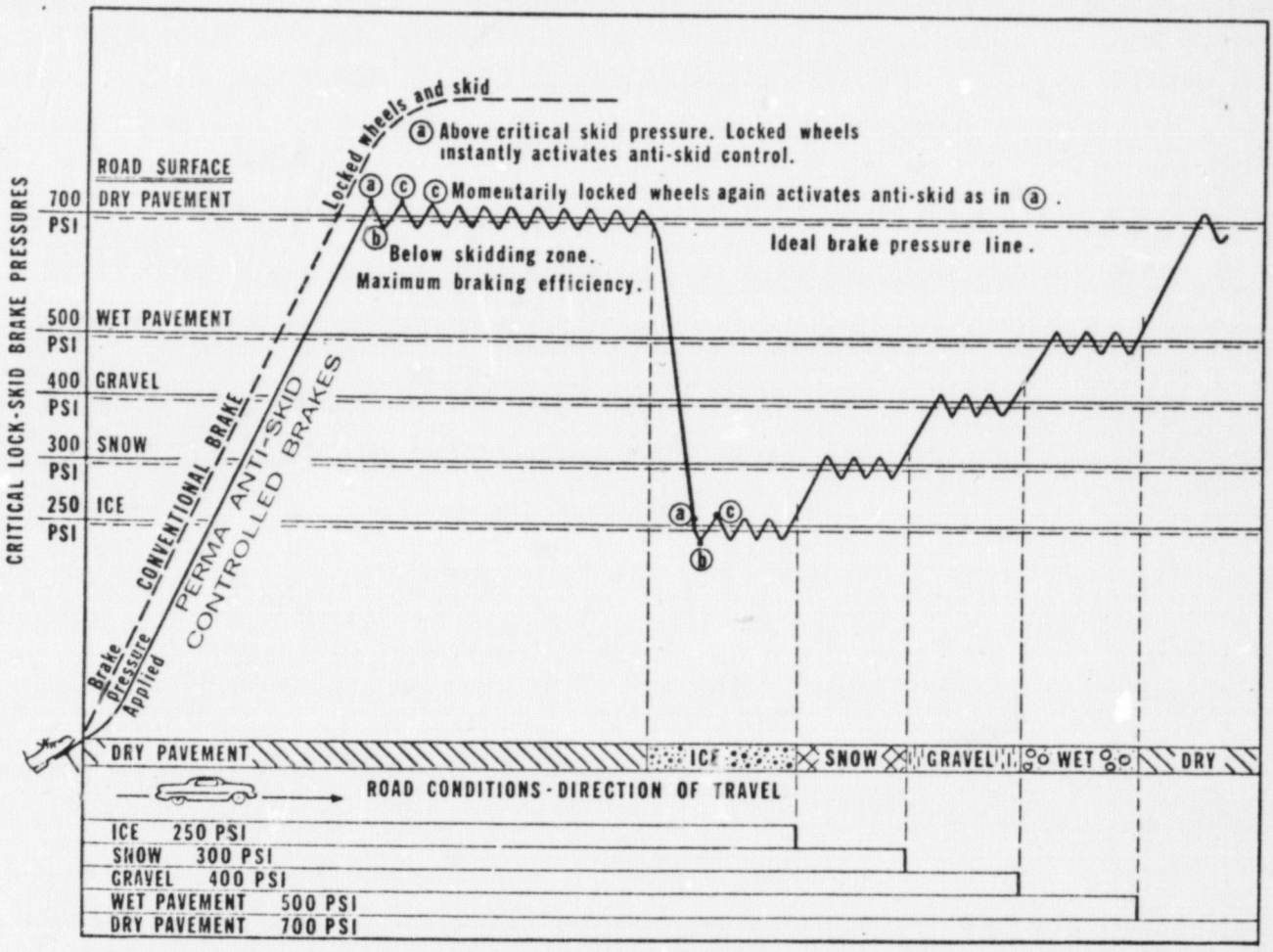


Universal Model B Perma Anti-Skid Unit Assembly shown is complete system—master brake cylinder (Upper left) is original equipment on vehicles and is not replaced.



Photograph above shows anti-skid unit disassembled. Metal has been replaced with "DuPont Delrin" which increased units reliability and life span. Precision requirements of various parts was not possible until metal was replaced with "Delrin".

GRAPHIC ILLUSTRATION OF ANTI-SKID SYSTEM



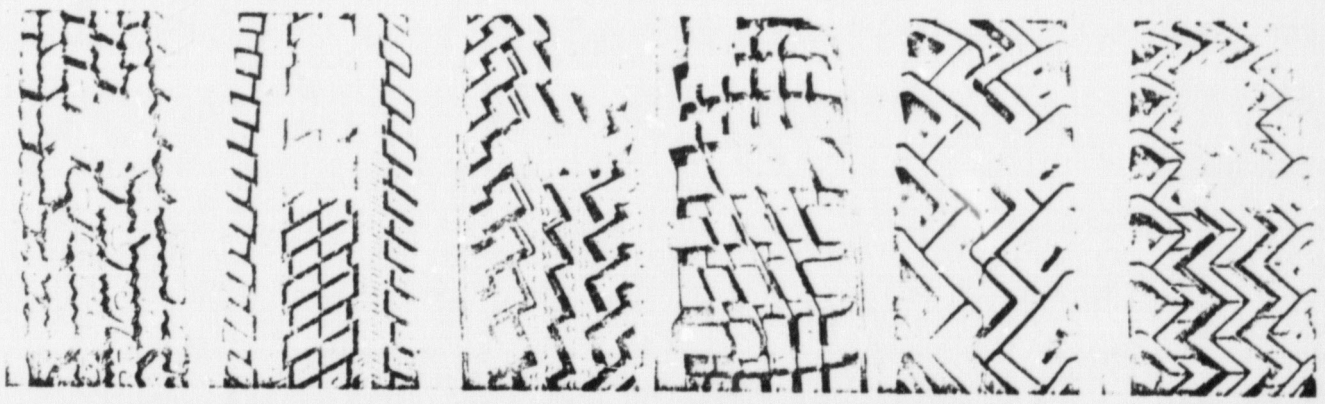
THE CHART ABOVE SHOWS A DOTTED LINE ILLUSTRATING THE LINE PRESSURE IN A CONVENTIONAL BRAKING SYSTEM WHEN HIGH BRAKE PEDAL FORCE IS APPLIED WHILE A VEHICLE IS ON A DRY, WET, GRAVEL, SNOW OR ICE COVERED SURFACE. IT CAN BE SEEN THAT WHEEL LOCK-UP OCCURS AT VARIOUS PRESSURES DEPENDENT UPON THE TYPE OF SURFACE THE VEHICLE IS ON AT THE TIME. THE PRESSURES SHOWN ARE THEORETICAL BUT ARE FAIRLY ACCURATE. THESE PRESSURES CAN ALSO BE RELATED TO THE COEFFICIENT OF FRICTION THAT EXISTS BETWEEN THE TIRE AND THE ROAD DEPENDING ON THE TYPE SURFACE AT THE TIME. FOR EXAMPLE, WHEEL LOCK-UP OCCURS AT 700 PSI LINE PRESSURE ON A DRY PAVEMENT WHICH INDICATES A .7 COEFFICIENT OF FRICTION WHILE LOCK-UP OCCURS WITH 250 PSI ON AN ICE COVERED SURFACE HAVING A .25 COEFFICIENT OF FRICTION.

THE IDEAL BRAKE SYSTEM IS ONE THAT WOULD RETARD WHEEL ROTATION UP TO A POINT JUST BEFORE THE WHEELS WOULD LOCK. THIS IS TERMED AN IMPENDING SKID CONDITION. THE SHORTEST POSSIBLE STOPPING DISTANCE WOULD BE OBTAINED FOR A GIVEN ROAD SURFACE WITH A MINIMUM OF POSSIBLE SKIDDING IF AN IMPENDING SKID CONDITION EXISTED.

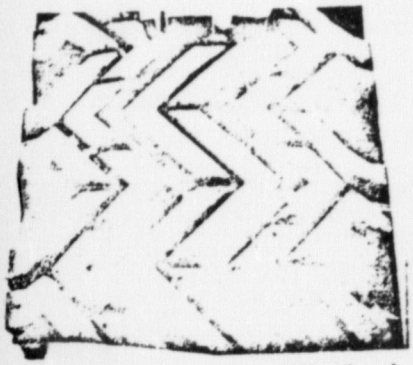
THE CREATION OF A PERFECT IMPENDING SKID CONDITION FOR AN AUTOMOBILE WOULD BE QUITE DIFFICULT DUE TO SUCH FACTORS AS WEIGHT TRANSFER, WHEEL LOADING CHANGES, ETC. THAT ARE PRESENT WHEN A VEHICLE IS MAKING AN EMERGENCY STOP. THE SYSTEM DISCUSSED IN THIS REPORT DOES HOWEVER CREATE SUCH A CONDITION FOR A LARGE PERCENTAGE OF THE DISTANCE OVER WHICH A VEHICLE IS BEING BRAKED WHICH ACCOUNTS FOR THE STOPPING DISTANCES SHOWN FOR THE DRY PAVEMENT TESTS MADE.

THIS SYSTEM COULD BE TERMED A CYCLE BRAKING SYSTEM THAT ALLOWS STEERING CONTROL WHILE BRINGING THE VEHICLE TO REST IN THE SHORTEST POSSIBLE STOPPING DISTANCES ON A NUMBER OF ROAD SURFACE CONDITIONS.

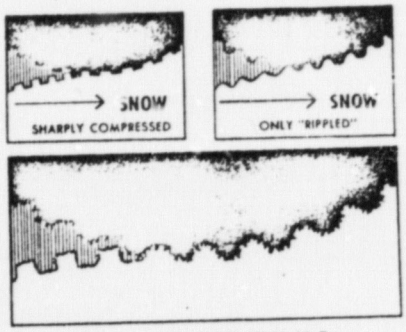
STOPPING ON WET, SNOW AND ICE COVERED ROADS



Illustrated above are 6 tread designs each of which rely on a mechanical function of gearing the tread edges into the snow to obtain traction. When edges of cleats, blocks or chevrons become worn, the traction ability of the tire is reduced. The stopability of the tire is also reduced because the original sharp edges of the tread cleats become rounded and slide over the snow covered surface instead of interlocking.

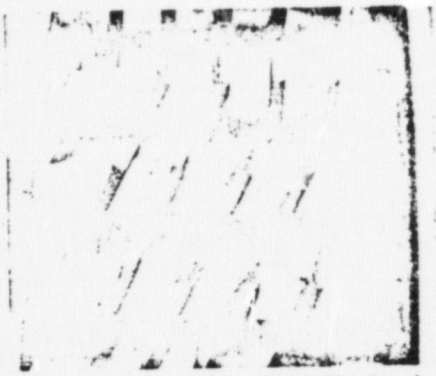


Actual worn tread edges after 2000 miles of use.

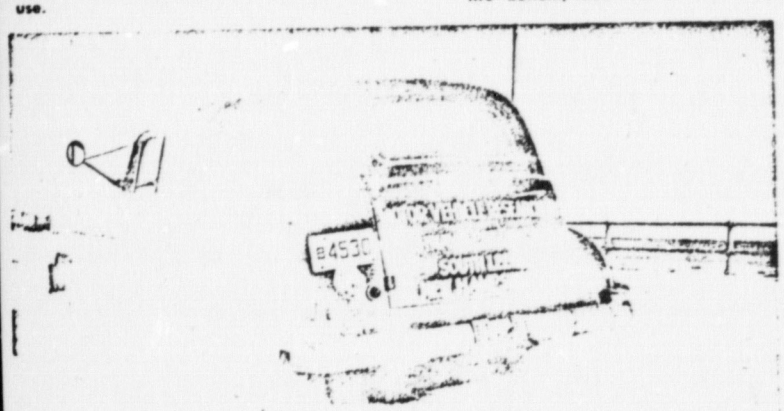


TEST OF SNOW TIRES BY M. V. R.

Illustrating new worn and ideal tread for snow traction—top left, new tread—top right, worn tire—bottom, ideal traction and stopping tread.



Actual worn tread edges after 1500 miles of use.



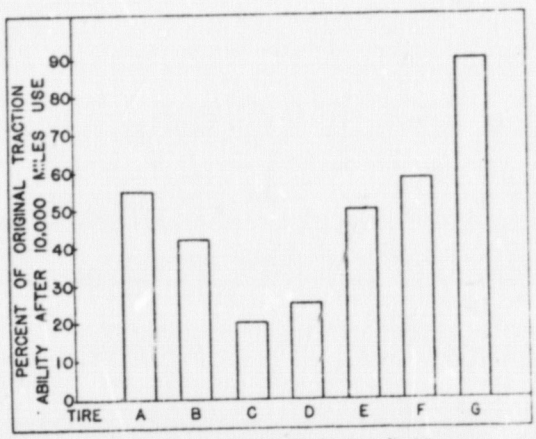


Chart above is calculated from other data obtained during tests. This chart shows the percent of original traction available for each brand of tires versus miles traveled. It can be noted that Tire G maintained a high degree of traction over the entire mileage range up to 10,000 miles. This is attributed to the renewable tread contact area produced by this unique process of impregnating the tread stock with sawdust.



Testing anti-skid system with wheels on snow.

Testing anti-skid system on ice

The true value of an anti-skid braking system is emphatically demonstrated when a driver attempts to stop under emergency conditions on a wet or snow covered road. With a conventional braking system and snow tires, a vehicle generally goes out of control when a driver fully applies the brakes. A popular method for preventing skidding on slippery surfaces is pumping the brake pedal. Such a method has real merit in preventing a skid but falls short of effectiveness in bringing the car to rest in a short distance. The reason for this is, the average driver is unable to pump fast enough and at the proper time as well as not being able to sense wheel lock-up and impending skidding intervals. In brief, the time interval is short and the average driver is too concerned with the danger confronting him to act with precision in pumping the brakes. Even under the most favorable conditions of stopping on a snow covered road, the distance to stop is long and usually not the shortest possible stopping distance.

Snow tires have no mysterious or exceptional qualities for providing traction and stoping in snow. They work on an almost pure mechanical basis, gearing the tread design into the snow to prevent wheel spin.

However, there is a distinct difference between the function of a snow tire when it is starting and stopping a car under emergency conditions. When the car is going forward under its own power there is some wheel slippage and almost constant tire rotation. This rotation prevents snow from filling in the tread to form a smooth surface and the edges of the tread design can act like a gear tooth train to cut into and push against the snow.

Under emergency stopping conditions with the wheels locked-up, the foot print of the tire becomes a smooth surface because the snow is compressed into the tread grooves and is not able to escape. When this occurs the tire acts somewhat like a skate and the car goes out of control.

An anti-skid device has merit in stopping a car on a snow covered road as it allows wheel rotation which prevents the snow from filling in the tread grooves. The tread edges then have an opportunity to act like a gear tooth train cutting into the snow surface and retarding the car faster.

A similar condition exists on a wet surface. Wheel rotation allows the water to be thrown out of the tread design and reduces the time a tire is sliding on a film of water.

When the tread edges of a tire become worn, the stopability of the tire is reduced and its traction ability is lessened. The anti-skid device has two basic functions that produce valuable safety conditions on wet or snow covered roads.

The first and most important of these functions is the ability of the device to prevent skidding when a locked wheel emergency stop is made. Also, the device creates almost constant wheel rotation during an emergency stop. This wheel rotation allows a driver to have steering control of his vehicle which in many cases will prevent a collision.

The second important function is keeping the tread grooves clean of water and snow through wheel rotation thus creating shorter stopping distances in most cases.

[PX 19, p 5]

(Continued from Page 5)

anti-skid system. The additional features of the system are equally important as the anti-skid feature.

I would like to sum up the overall features in the order of their importance.

(1) The system, when applied to a standard passenger car, shows its ability to stop a vehicle in a shorter distance by producing an impending skid condition throughout the total stopping distance. The following figures have been recorded:

(a) From 60 MPH it required 171.9 feet to stop with all wheels locked and 193.2 feet when the driver pumped the brakes. It required 156.3 feet to stop using the modified system.

These figures are the average of 3 test runs in each area. Tire and brake engineers are most happy to achieve a 1% improvement in performance from time to time. These tests show almost 15% in improvement.

(b) From 50 MPH the figures were 122.5 feet for locked wheels, 128.11 for fast pumping and 112.0 feet using the modified system.

(c) From 40 MPH the figures are 83.0 feet, 89.3 feet and 72.0 feet.

(d) From 30 MPH, 50.1 feet, 55.11 feet and 46.0 feet. Such improvement in stopping distances are remarkable.

(2) Tests of stopping to avoid striking an object were conducted using locked wheels and with the device. In 24 tests, the vehicle equipped with the system was steered away from the object and stopped in a shorter distance. In this area, such steering control is of extreme value.

Our accident investigations reveal that many persons could have avoided accidents if their vehicles were equipped with such a system.

Bear in mind, that under locked wheel stopping conditions, you cannot steer a vehicle. It will continue in a line it is committed when the brakes lock up the wheels.

This is then an important feature in safety.

Frankly, our tests show this system to be a new and unique safety braking system that is long overdue in the motoring world. We are following these tests with others on wet, sand covered and icy roads. We anticipate equally good results.

This system can be called the braking system of the future.

PRINCIPLE OF OPERATION "MECHANICAL BRAIN"

The master cylinder is the reservoir from which fluid is delivered whenever the brake pedal is depressed. The anti-skid unit is the device which prevents an excess of pressure from reaching the wheels, which would cause a skid.

The anti-skid unit employs a gyro governor that always rotates in direct relationship to the vehicle wheels and speed of the vehicle.

The governor does two jobs; it senses a locking wheel condition by the loss of its own rotation in relationship to vehicle speed and transfers this message to the electrical solenoid and armature to allow the unit to instantly go into operation.

Secondly, it controls the operation of the Perma-Vac through its valve mechanism. This valve under control of governor position modulates fluid pressures.

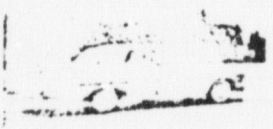
EXAMPLE: To start the cycle of the anti-skid in operation, we will assume a vehicle speed of 50 MPH, bearing in mind that the more slippery the road surface, the less pressure is required to lock the wheels, which causes the skid. On a dry road 700 PSI of fluid pressure would lock the wheels; the governor must relieve just enough pressure to allow the wheels to turn, yet stop in the shortest possible distance utilizing all road-to-tire friction available. Thus, the 700 PSI would be reduced to 650 to accomplish this end. If during this type of dry pavement stop employing 650 PSI, the vehicle approached an icy patch, the governor would automatically reduce the brake pressure to compensate for the more slippery suface.

The unit also employs fail-safe features for added driver protection.

The "Mechanical Brain" is a small, but complex mechanism featuring a gyro govenor, which is connected directly into the brake system and transmission. The unit is able to sense in a split second the decrease in wheel revolutions that accompanies a "wheel lock" condition when brakes are jammed on a slippery surface. The anti-skid unit reacts immediately to sufficiently relieve brake pressure while imparting the pumping action which allows the driver to retain steering contol during deceleration.

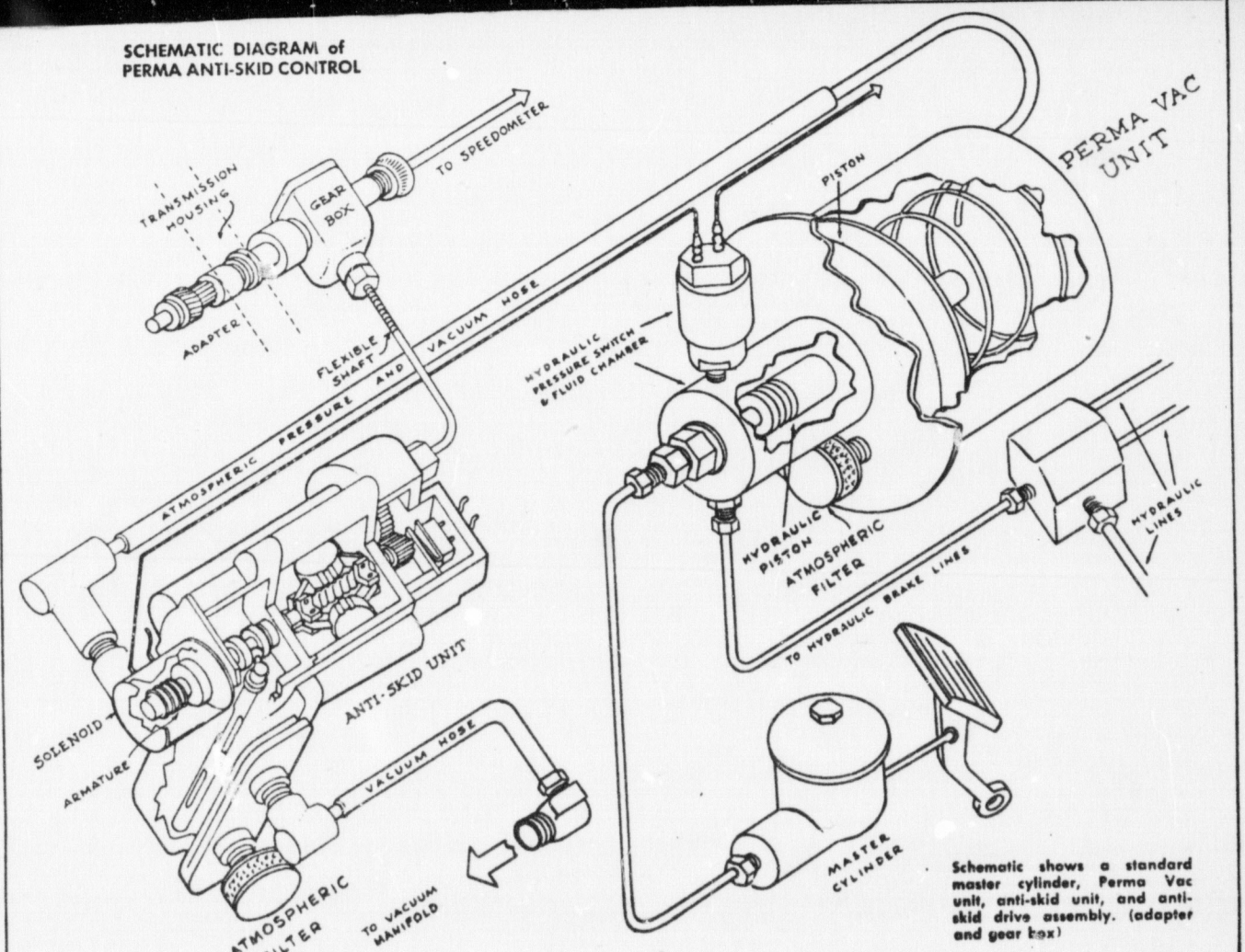


A low speed emergency stop—note smoke from tires and wheel chatter—standard braking system.



A low speed emergency stop with new braking system—no wheel chatter or tire smoke is seen.

SCHEMATIC DIAGRAM of PERMA ANTI-SKID CONTROL



Schematic shows a standard master cylinder, Perma Vac unit, anti-skid unit, and anti-skid drive assembly. (adapter and gear box)



Scene of M. V. R. wet surface testing

WET PAVEMENT TESTS

Motor Vehicle Research wet pavement tests using an anti-skid device versus a standard braking system closely parallel similar tests by the "Department of Scientific and Industrial Research of the British Government"

Stopping an automobile on a wet pavement is of prime importance to every motorist. Brakes are usually adequate for this purpose when properly applied bu. sometimes emergencies arise in which a car must be stopped in a short distance. When brakes are used under emergency conditions on wet pavements, the wheels of the vehicle usually lock up and the vehicle then slides on the tires to come to rest. Under such conditions, the only means of stopping the vehicle is the friction between the roadway and the tires. The available friction under most wet conditions is usually low even with the best tire designed for these conditions.

A great number of tire design features are used in an effort to obtain high friction values between tires and wet surfaces. Most of these are of value with some tires performing better than others. Two things are of vital importance under locked-wheel, wet pavement vehicle stopping conditions and both deal with tire design. One is the type of tread design where the tire "wipes the road" and egress for the water between the tire and road is available. The second is a mechanical function in the tread design consisting of a great number of tread edges or sipes that act like scrapers sliding on the pavement absorbing the energy of the vehicle. Both are of equal importance because they are interdependent.

The wiping action of a properly designed tire tread functions quite satisfactorily during emergency stopping at low speeds but becomes ineffective as the speed rises. For example: a tire that allows water egress through rib design and also provides a wiping action at 20 m.p.h. might become ineffective at 30 m.p.h. and at 40 m.p.h. the tire will ride on a film of water under locked wheel conditions. The inclusion of an anti-skid device in the braking system allows wheel rotation which causes water egress to be of a constant nature and water retention in the tread grooves would be minimum. Such a system then becomes valuable as shorter stopping distances can be obtained.

There is ample evidence of the effectiveness of an anti-skid system in wet pavement stopping. One outstanding report can be found in a text book titled "Research on Road Safety" published by Her Majesty's Stationery Office, York House, Kingsway, London W. C. 2. The test of an anti-skid device was conducted by the Road Research Laboratory, an agency of the British Government and reported in this text. The findings of this organization were very similar to those of Motor Vehicle Research of New Hampshire testing the Perma Anti-skid Braking System.

In the Road Research Laboratory Report the minimum braking distances obtained in repeated tests from speeds of 30 and 45 m.p.h., with and without the anti-skid units in use, for a number of surfaces when wet, showed the braking distances were less when the anti-skid unit was used compared to locked wheel tests from the same speeds.

The reduction in braking distances depended on the nature of the road surface and on the initial speed. The greatest improvement was obtained from the higher speeds on the more slippery surfaces.

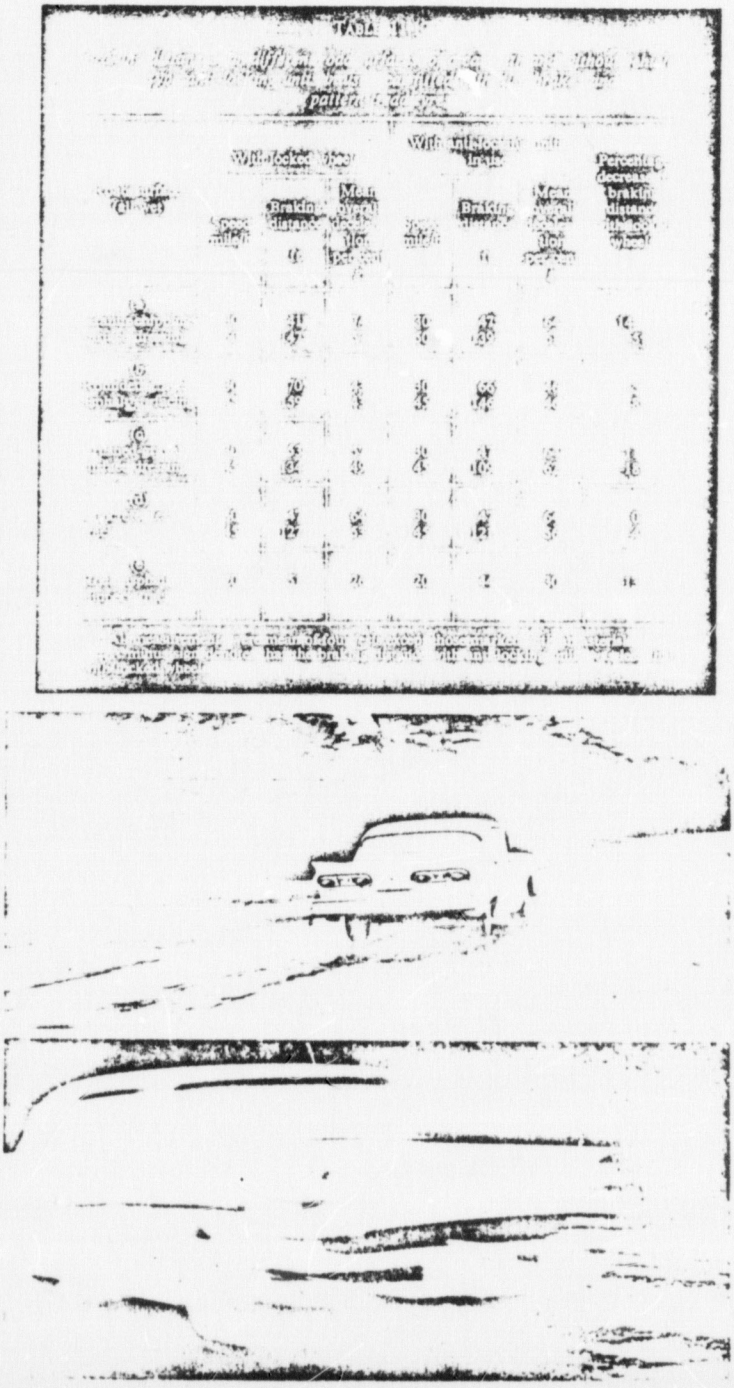
They also reported excellent steering control of the vehicles in a slalom course test.

Table 11.17 from the text is reproduced here. The data contained in this table confirms the M. V. R. tests and shows a passenger car can be stopped in a shorter distance using an anti-skid braking system on a wet slippery road.

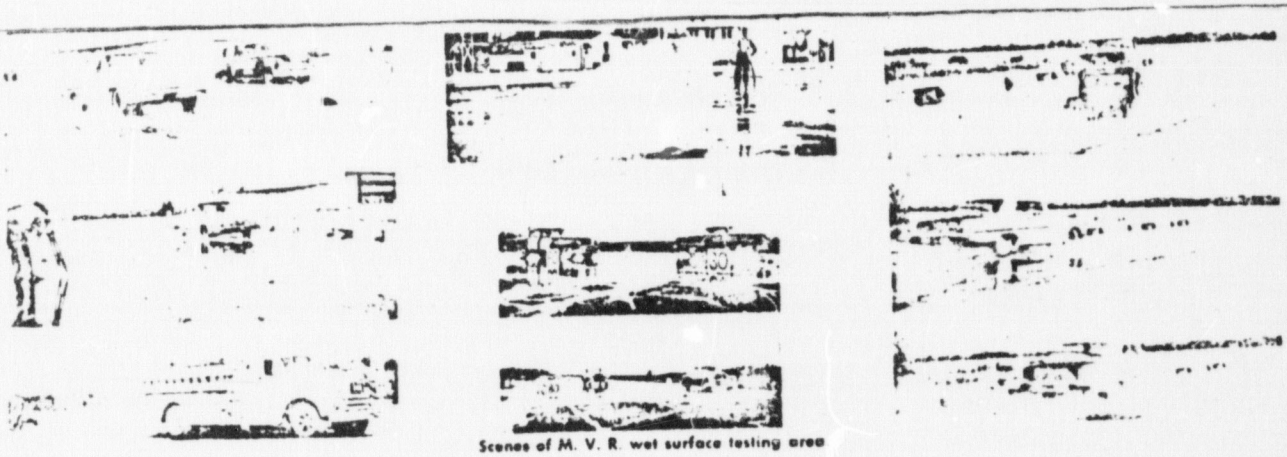
The magnitude of the problem of skidding on wet roads is clearly shown in the text described above (Page 490). In this summary 27% of all accidents on wet roads involved skidding and 73% of all accidents on snow or ice covered roads involved skidding. It can be concluded from such data that an anti-skid braking system is extremely valuable and the Road Research Laboratory of England (page 417 of the text book) states that an anti-skid braking system has distinct advantages over conventional braking systems.

The M. V. R. wet pavement tests were conducted on an asphalt airport runway which was constantly flooded with water to produce a reasonably uniform wet suface.

Water tank trucks with pumps were used to supply spray hoses with water at a rate up to 60 gallons per minute. After the pavement became saturated a constant water spray was applied to maintain a uniform wet surface.

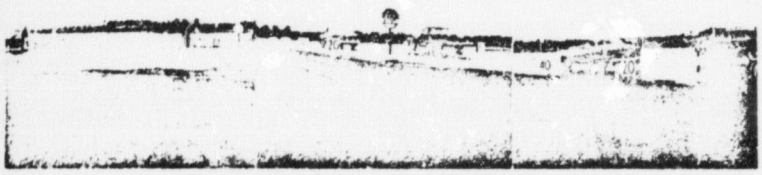


Upper wet pavement test—Lower—note tire deformation.



Scenes of M. V. R. wet surface testing area

[PX 10, p 7]



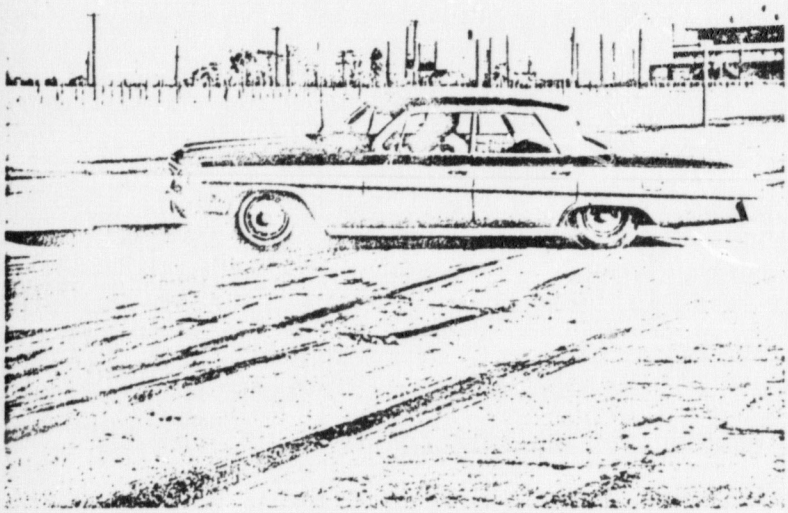
Scenes of M. V. R. wet surface testing area.

Vehicle behavior emergency braking tests with and without anti-skid system from 70 and 80 miles per hour.

A series of high speed stops were made from 70 and 80 miles per hour to examine vehicle behavior characteristics with and without the anti-skid device.

These tests confirm conclusively that the value of the anti-skid system increases as the speed of the car increases. The system produces shorter stopping distances from speeds of 20-30, 40-50 and 60 m.p.h. than the conventional system. The added and important feature is the availability of steering control when the anti-skid system is used.

Test drivers reported no problems stopping from 70 and 80 m.p.h. using the system while they stated they were forced to release the brakes when the same tests were tried with the conventional braking system.



Without anti-skid system car rotates in out-of-control manner turning end for end.



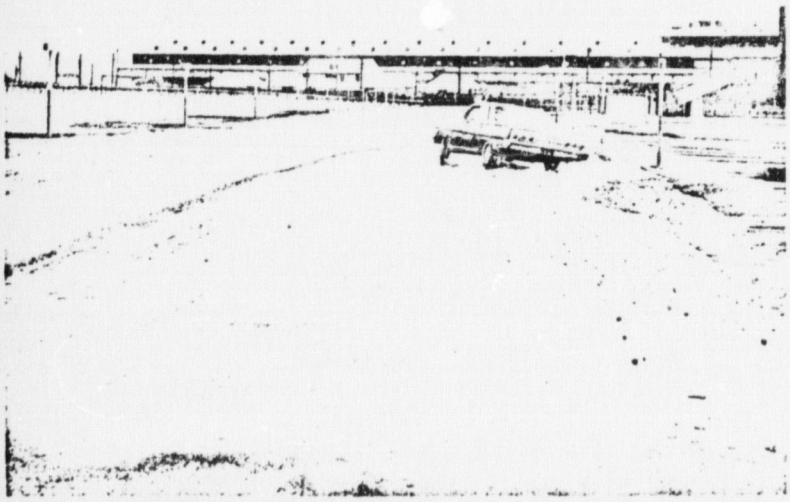
80 m.p.h. stop with anti-skid system. Note interruptions in tire marks and straight stop.



Photograph shows how anti-skid system applies and releases brakes while holding impending skid at regular intervals.



80 m.p.h. stop with locked wheels using conventional braking system. Tires are smoking and driver was forced to release brake to prevent over-turning.



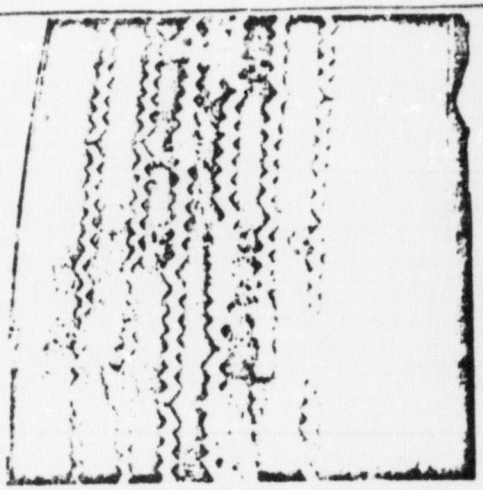
Out-of-control wet-surface stop. Note driver has turned front wheels in attempt to correct. Lack of anti-skid system caused skid and steering to correct proved futile.



Open tread design of snow tire which prevents snow accumulation. Tread bars can cut into snow.



Test stop on soup covered wet roadway with anti-skid system.
Car is under control and stop is straight.



Tread grooves filled in with snow to form an unbroken surface.
Tire will slip as no tread edges are available to bite into snow.

[PX 19, p 8]

FIELD TESTS—STOPPING ON DRY PAVEMENT

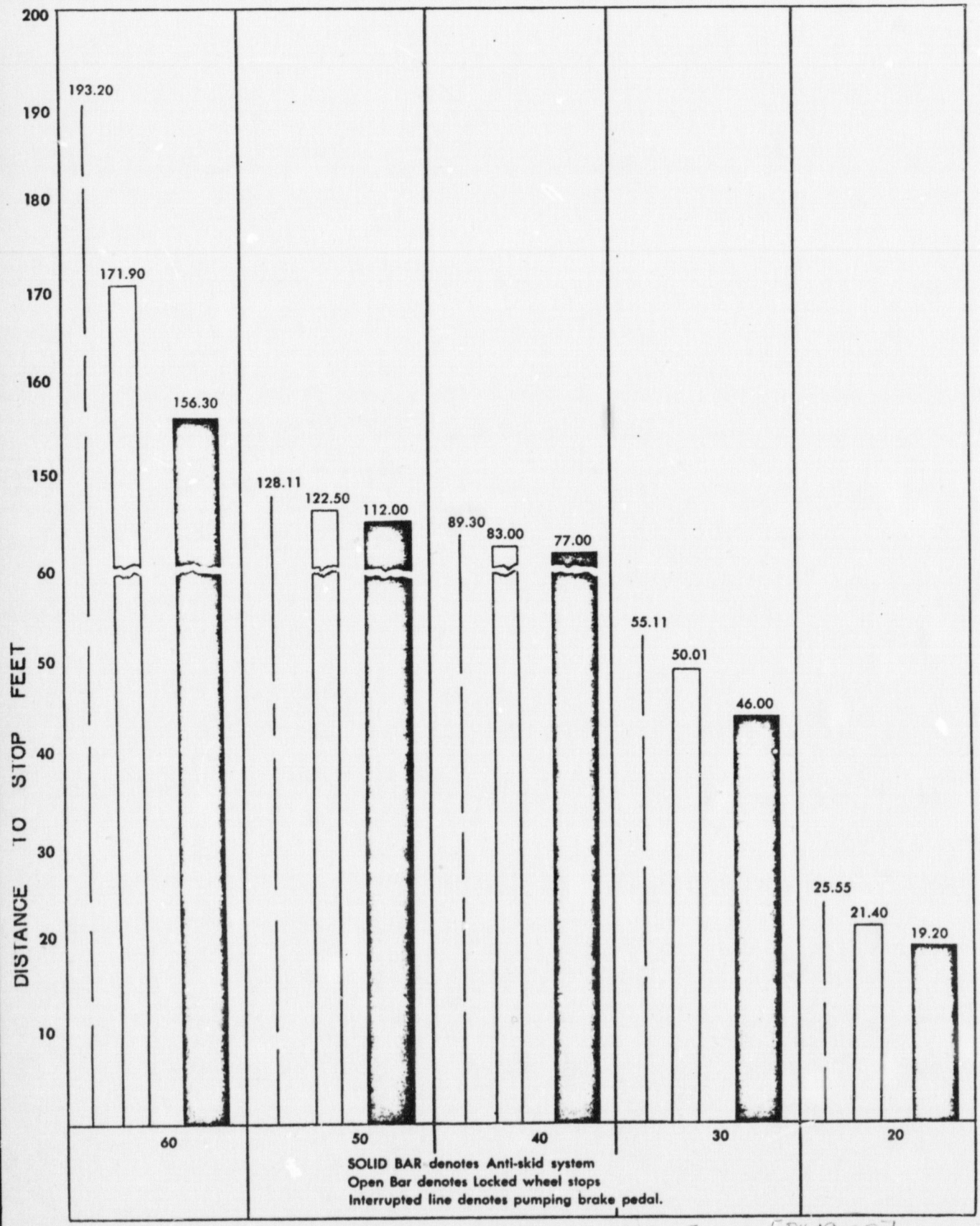
STATISTICAL DATA OF TESTS AT A GLANCE

The effectiveness of any device can only be determined by actual controlled tests under use conditions. In the case of determining the stopability of a vehicle, a number of factors must be known. One factor consists of the determination of the vehicle speed at the moment the brakes are applied. The use of a vehicle speedometer indication for determining this factor would lead to error due to a number of controls such as angle of viewing the speedometer pointer relative to the numerals on the dial and the driver's interpretation of this relationship. Another source of error would be the accuracy of speedometer system itself when coupled to the drive train of the vehicle.

The most accurate method for determining a vehicles speed was developed by Motor Vehicle Research during the early 1950's. This is known as the vehicle time displacement method using syncronized high speed motion picture cameras mounted on the vehicle to view a syncronized electric timer and a reference fence positioned on the roadway. (See photographic record section)

This method produced a film record of the vehicle speed before brake application. The moment the brakes were applied, the deceleration rate in milliseconds of time and the distance the vehicle traveled in this time.

The bar graph below shows the effectiveness of the anti-skid system in stopping the vehicle in a shorter distance from speeds of 20—30—40—50 and 60 miles per hour. Compared to locked wheel stops and brake pedal pumping, the anti-skid provided a superior braking system by stopping the car in shorter distances. This graph shows the average of 3 stops from each speed with and without the device and pumping the brake pedal.



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GENERAL DISCUSSION OF DRY PAVEMENT TESTS

The shorter stopping distances found using the modified system during dry pavement tests can be accounted for in the afore mentioned factors in several combinations.

It should also be pointed out that the mechanical action between the tire and the pavement is normally higher for a relatively cold polymer vs. a hot polymer as the force required to shear the cold polymer is higher than the force required to shear a polymer close to or above the melting point. This high resistance to mechanical shear is also of value in creating a high average coefficient of friction.

It was also noted from a study of the normal motion picture film and the time displacement data that vehicle deceleration was of a relatively constant nature even during the cycle period of the modified system. Brake torque was constantly present and this was determined from a study of the rotational factor of a free rolling wheel over a distance versus a wheel with retarded rotation relative to the normal rotation.

While it is our opinion that all such factors were not deliberately designed into the system, it is a coincidence that such factors blend to create better vehicle stopping under emergency conditions.

It was also noted that the difference in stopping distances between the anti-skid braking system vs. the locked wheel stops became greater as the speeds increased. This was found in both the dry and wet pavement tests.

The stopping distances measured by the distance between detonator shell markings and vehicle time-displacement measured with high speed motion pictures indicate the existance of an impending skid condition during panic stops.

The magnitude of the differences in stopping distances using the normal braking system and the modified system is of a significant nature in favor of the modified system.

A study of normal speed motion picture film recording the rotational action of the vehicle wheels during test stops involving the modified system shows an impending skid existing for long time intervals during the total stopping times.

It was noted that the totally locked wheel time during each test stop, was comparatively short while the impending skid time was relatively long.

It was also noted that during the locked wheel tests, the tire noise level was constant and high while with the modified system the level was low and intermittent.

The normal speed motion picture film also showed a constant braking action by the vehicles wheels which indicates the response action of the modified system was such as to cancel total brake shoe drum inter-actions usually associated with the "pumping of the brakes" during vehicle braking. The action of the vehicle wheels seen in the motion pictures indicates that the brake shoe pressures acting on the drum interface are not reduced to zero at any time.

It is theorized that the servo action of the braking system design is not totally absent during any cycle of the modified braking system, and this system produced up to eleven cycles during some test stops. It is thought that this servo action prevents the brake shoe from completing a cycle of contact/no contact to the brake drum thus holding braking torque to the wheel.

While brake line pressures may be reduced to zero several times during each stop using the modified system, the response characteristics of the mechanical linkage of the brake assembly may be too slow to allow a complete cycle of contact/no contact of the lining and drum when brake line pressures are changed from zero to several hundred p s i in milliseconds and repeated several times during one second.

An additional factor concerning the vehicle tires can contribute to the higher average coefficient of friction factor found when the modified system is used.

It is an established fact that tire tread footprints reach temperatures beyond the melting point of the polymer when a wheel is sliding on a pavement. Under such conditions, the coefficient of friction curve for a given distance has a high value initially and this value is reduced to a marked degree as the stopping distance is lengthened.

Presenting a new tread stock area to the pavement several times during one stop prevents temperature elevation of the polymer thus reducing the lubricating action of a melting tread stock and assists in maintaining a high coefficient of friction average.

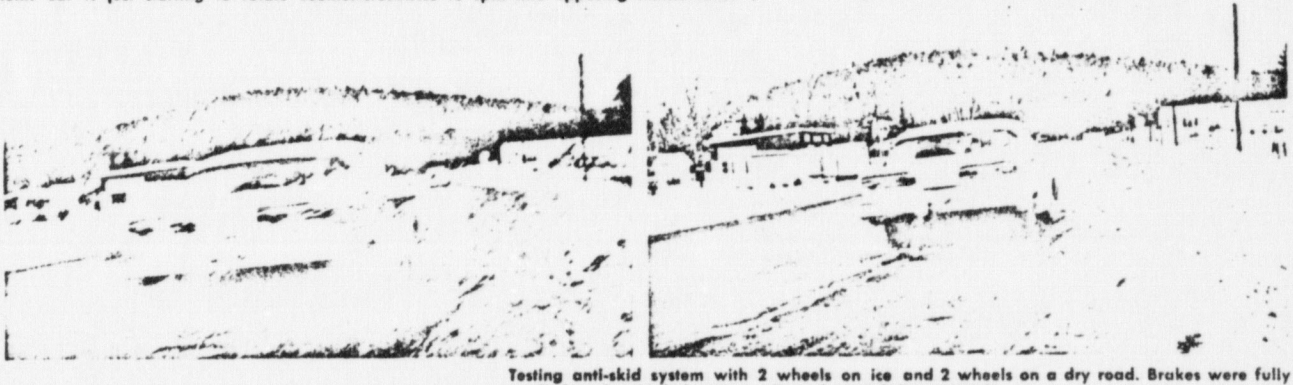
Tests with 2 Wheels on Wet Snow or Ice and 2 Wheels on a Dry Surface

Motorists often encounter situations that require emergency stopping when the two right or left side wheels are on wet or icy surfaces and the opposite wheels are on a dry surface. In situations of this type, the wheels on the wet or icy surface lock up and slide while the opposite wheels retard their side of the vehicle faster. The vehicle then goes into a skid rotating around the wheels that are on the dry road resulting in an out-of-control situation.

A vehicle equipped with an anti-skid braking system does not go out of control because the system senses the wheels that lock-up and releases them to a point of impending skid. The vehicle is controlled under these conditions.

Tests were conducted on roads having ice covered shoulders and dry clean travel lanes. The vehicles were driven with the two right side wheels on the ice covered shoulder while the opposite wheels were on the dry travel lane. The brakes were applied fully and held until the vehicle came to rest. Without the anti-skid system in operation, the car went out of control into opposing traffic lane. With the system, the car continued in a relatively straight line coming to a controlled stop.

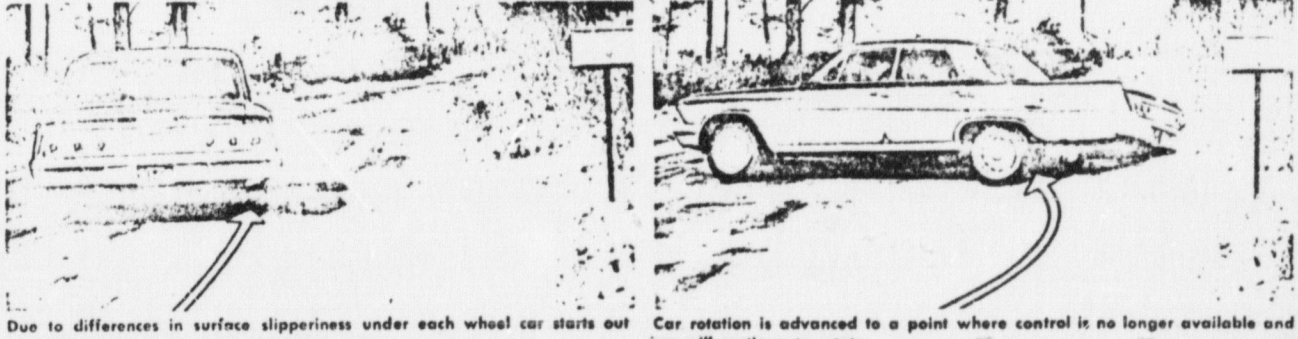
Testing conventional braking system with 2 wheels on ice and opposite wheels on dry pavement. Car is just starting to rotate counter-clockwise to spin into opposing traffic lane.



Testing anti-skid system with 2 wheels on ice and 2 wheels on a dry road. Brakes were fully applied and anti-skid system allowed straightline control.



Testing anti-skid system with two wheels on snow and 2 wheels on dry pavement. Vehicle control was satisfactory. Brakes were fully applied.



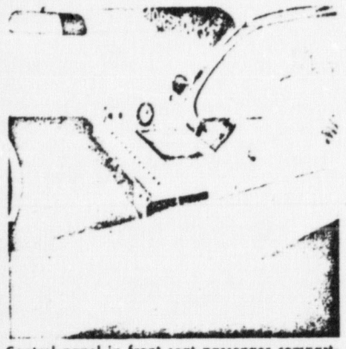
Due to differences in surface slipperiness under each wheel car starts out of control with a conventional braking system.

Car rotation is advanced to a point where control is no longer available and car will continue to rotate.

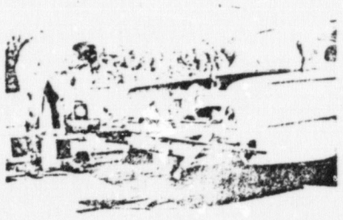
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[PX 19, p10]

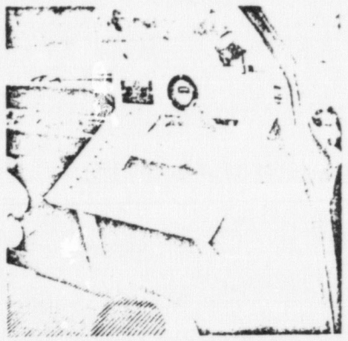
PHOTOGRAPHIC RECORD OF TEST CAR AND INSTRUMENTATION



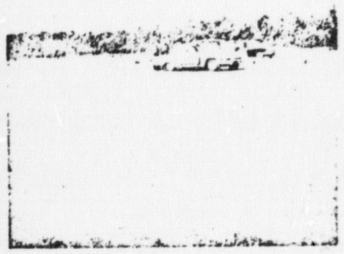
Control panel in front seat passenger compartment.



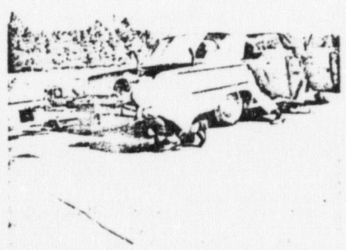
Instruments mounted on outrigger frame.



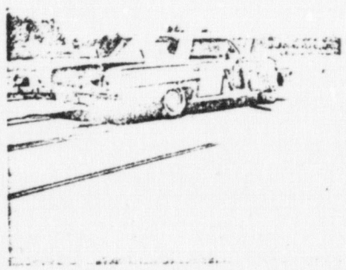
Switches on left of panel control timers, cameras and detonators.



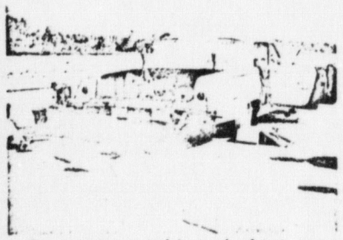
Test site in Maine.



Technicians aiming High Speed Motion Picture Camera.



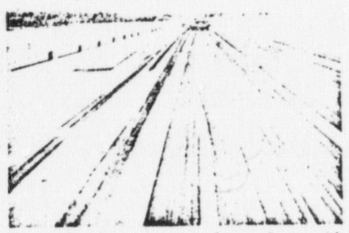
Test car after stop.



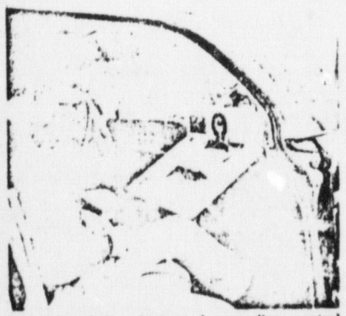
Generators mounted in trunk of test car.



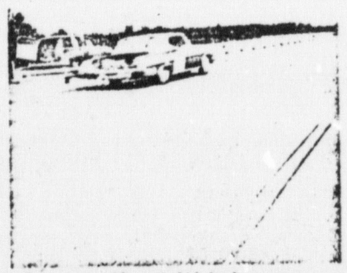
Tire markings to determine wheel rotation through High Speed Motion Pictures



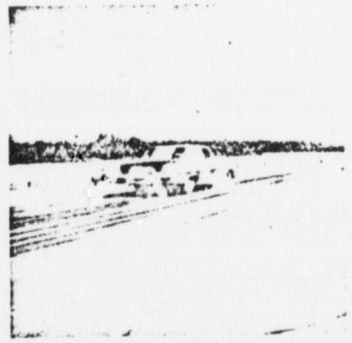
Tire marks after one series of tests—Fence with numeral on left is for camera record.



Technician being instructed regarding control



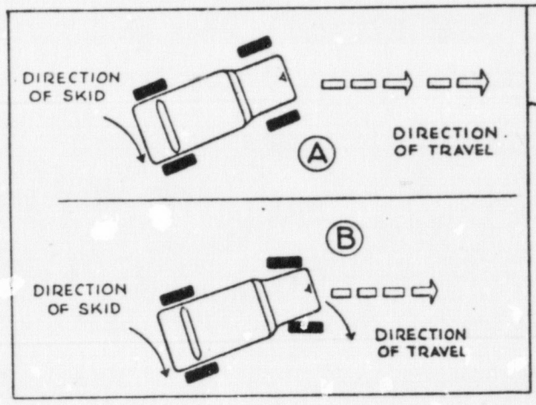
Test stop with anti-skid brake system.



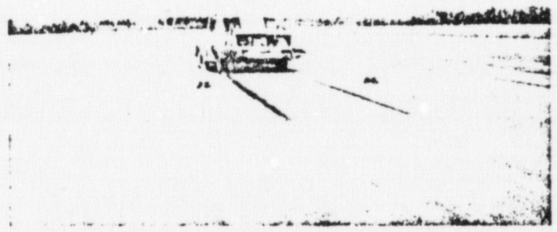
Test stop without anti-skid brake system.

STEERING CONTROL WITH ANTI-SKID SYSTEM A MEANS OF SKID PREVENTION

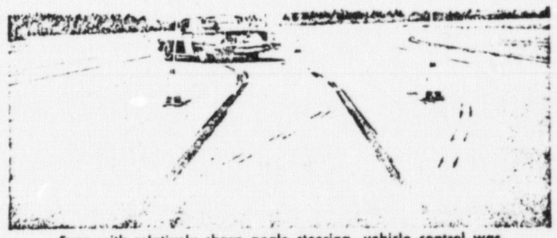
There are several distinct advantages present when an anti-skid braking system is used. The first and most important of which is the prevention of skidding through controlled braking. The diagram shown here illustrates the method for correcting a skid after it is initiated. This method has merit and should be part of every motorist's driving skills.



Drawing "A" shows the attitude of a car in a skid. The arrow shows the direction the rear wheels are going. In this case, the rear wheels will continue to go to the right unless the front wheels are turned as illustrated in diagram "B".



Photograph shows how vehicle could be steered while under emergency braking using the anti-skid device.



Even with relatively sharp angle steering, vehicle control was possible while shorter stopping distances were found when the anti-skid unit was used.

Any application of the brakes during a skid when a vehicle is not equipped with an anti-skid system and the front wheels are not turned in the direction of the skid, will cause the skid to become more violent.

It then appears that an extremely valuable safety factor can be gained with an anti-skid system. This factor is the prevention of a skid rather than attempting to correct a skid when it occurs.

A series of tests to study the steering control without the anti-skid system have been described, in part, on the opposite page. The photographic record of the action of the car equipped with this system is shown here.

It is also of interest to note that the car could be stopped in a shorter distance from the pylon as well as not impacting it when the anti-skid system was used. In both tests, with and without the anti-skid system in operation, the driver applied the brakes fully and maintained them fully applied until the vehicle came to rest. With the anti-skid system being used, a driver faced with an emergency situation has an automatic control working to reduce the danger of a collision. The only requirement of the driver is to apply the brakes fully and steer the car away from the object. Without the anti-skid system he would be in trouble and could not avoid a collision.

Another very important factor of benefit to a car owner is the ability of the system to sense unbalanced braking during an emergency stop and compensate for such a condition by reducing the braking force on any wheel that locks up. This is a most important area of benefit that the anti-skid system provides. Brake systems become unbalanced from lack of proper maintenance or from defects in the brakes on one wheel of an axle.

Many car owners are unaware of brake unbalance that exists in their vehicle because they are not continually involved in emergency stopping to test brake balance.

The unbalance usually becomes apparent when an emergency situation arises and in such cases it is too late to correct before a collision or out of control accident occurs. In such cases, the lack of directional stability that is present is the controlling factor causing the accident.

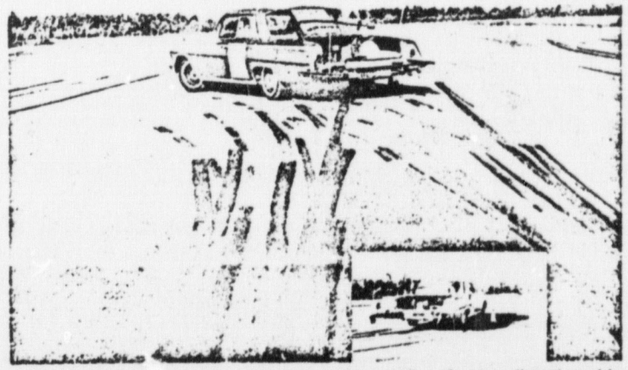
In the case of commercial vehicles, an anti-skid system would provide a valuable safety factor in overcoming the problems of inadequate braking on all wheels when these commercial vehicles or trucks are operating without pay loads.

Most commercial vehicles are equipped with brakes adjusted to a certain "K" factor and this is usually calculated to include a full payload. When such vehicles are braking in an emergency situation while unloaded, they will generally go out of control. If an anti-skid system was fitted only to the rear wheels it would overcome the difficulty of providing adequate braking on all wheels when the vehicle is loaded and prevent rear wheel lock-up when the load was removed.

Differences in loads carried by individual wheels, particularly in the case of commercial vehicles and the ratio of the applied braking effort between the front and rear wheels may effect the distance in which a vehicle can be stopped and loads can have an important influence on the behavior of the vehicle when full braking is utilized. Skidding and loss of directional control can be experienced even when brakes are in good condition and adjustment is proper. Front wheel unbalance can increase directional control difficulties.

It can be therefore understood that an anti-skid system has exceptional value in enabling an operator to control his vehicle.

Road super-elevations can influence wheel loadings in some cases. In such cases, an anti-skid device, detecting wheel rotation characteristics and controlling rotation, contributes to safety.



Photograph shows the impending skid condition that exists shows cycling of system. (Insert) precision steering was possible

E-142 [PX 19, PIZ]

Photograph above shows anti-skid unit disassembled. Metal has been replaced with "DuPont Delrin" which increased units reliability and life span. Precision requirements of various parts was not possible until metal was replaced with "Delrin".

STEERING CONTROL PERFORMANCE TEST

An acceptable anti-skid braking system provides a most valuable safety factor by allowing steering control during emergency stopping

A large number of accidents could be avoided if the vehicles involved were equipped with anti-skid braking systems. It is not uncommon to see a set of relatively long tire marks that terminate at a point where debris is scattered over the road surface and two vehicles resting close by in a state of twisted metal. Further examination of the area would show ample room for passing around the debris. The question then arises as to why the vehicle or vehicles did not steer away from each other and avoid the collision. The answer is relatively simple. First, the long tire skid marks show that the vehicle wheels were not rotating but were "locked-up". When this condition exists the operator of a vehicle has no steering control. He can turn the steering wheel fully left or right which will change the vehicle wheel direction but the car will continue in the same path it was committed to just as the brakes were fully applied. If he was sufficiently skillful and could "pump the brakes" fast enough and also maintain an impending skid condition between the tires and the road, he could steer the car and also bring it to a stop in a relatively short distance.

Unfortunately, drivers are not able to even approach such skills, therefore a braking system that will perform such a function is most valuable. An acceptable anti-skid braking system can satisfy all the requirements for providing steering control in emergency stopping and bring a vehicle to rest in the shortest possible stopping distance for the road condition.

Motor Vehicle Research of New Hampshire conducted tests to evaluate the anti-skid system under discussion to determine its ability to maintain a high deceleration rate and at the same time allow wheel rotation to provide steering control. These tests were conducted on a road and 2 human operators drove the vehicles.

Motion picture cameras recorded the movement of the front wheels of the vehicle while the stopping distance and time was recorded via high speed timers, detonators and cameras mounted on an outrigger frame located on the rear of the vehicle.

A triangle course was marked by 3 pylon and wood uprights. (See photograph).

The driver was instructed to drive the car between two pylons spaced 12 feet apart and apply the brakes fully at a moment when the front of the vehicle was opposite these pylons and hold the brakes applied until the car came to rest. He was further instructed to steer the vehicle to avoid striking a third pylon positioned 75 feet between and beyond the first 2 pylons. (See photograph A).

Six test stops from 40 m.p.h. were made without the anti-skid system in operation. In these tests the drivers turned the steering wheel hard to the left in an effort to avoid striking pylon #3 but was unsuccessful. He was able to change the angle of the front wheels by turning the steering wheel but the vehicle continued to go straight ahead to impact the pylons. (See photographs).

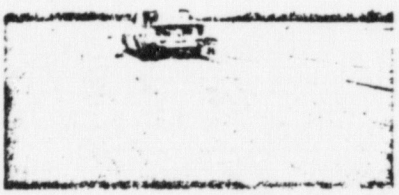
Using the same conditions and the anti-skid system in operation, the driver was able to steer away from striking the pylon and stopped in a 12% shorter distance.

These tests confirm similar tests conducted with an aircraft type anti-skid system and reported in the book "Research on Road Safety" published by Road Research Laboratory at Harmondsworth, Middlesex, England. (Pages 416-419). The organization is an agency of the British Government.

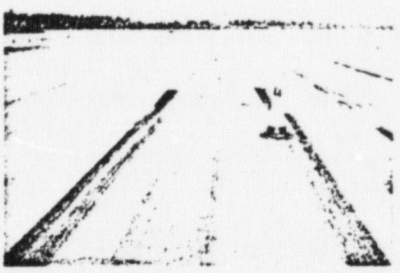
As stated previously, steering control during emergency braking is an important safety factor and can be the means of preventing vehicle collisions. This anti-skid device provides such steering control.



Pylons placed to produce triangle course.



Test vehicle entering course.



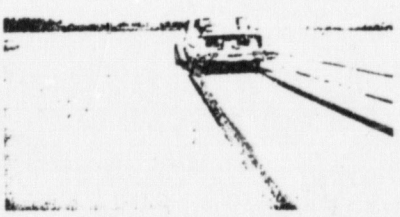
All wheels are locked without anti-skid device and car continues straight ahead to impact pylon.



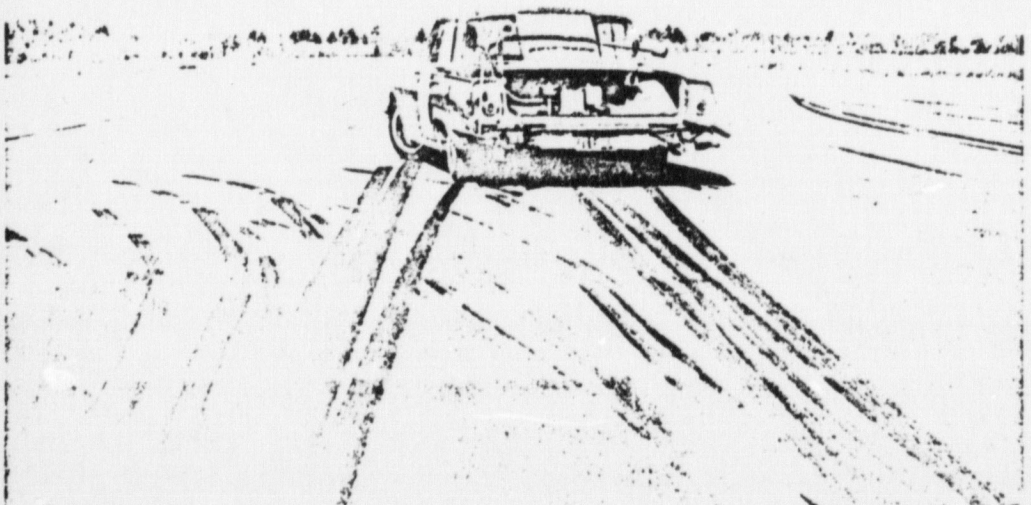
Course of car without anti-skid device. Impacted pylon has been replaced to show conditions.



Extent driver turned wheels to unsuccessfully steer car without anti-skid system.



Note wide front wheel tracks caused by operator trying to steer without anti-skid device.



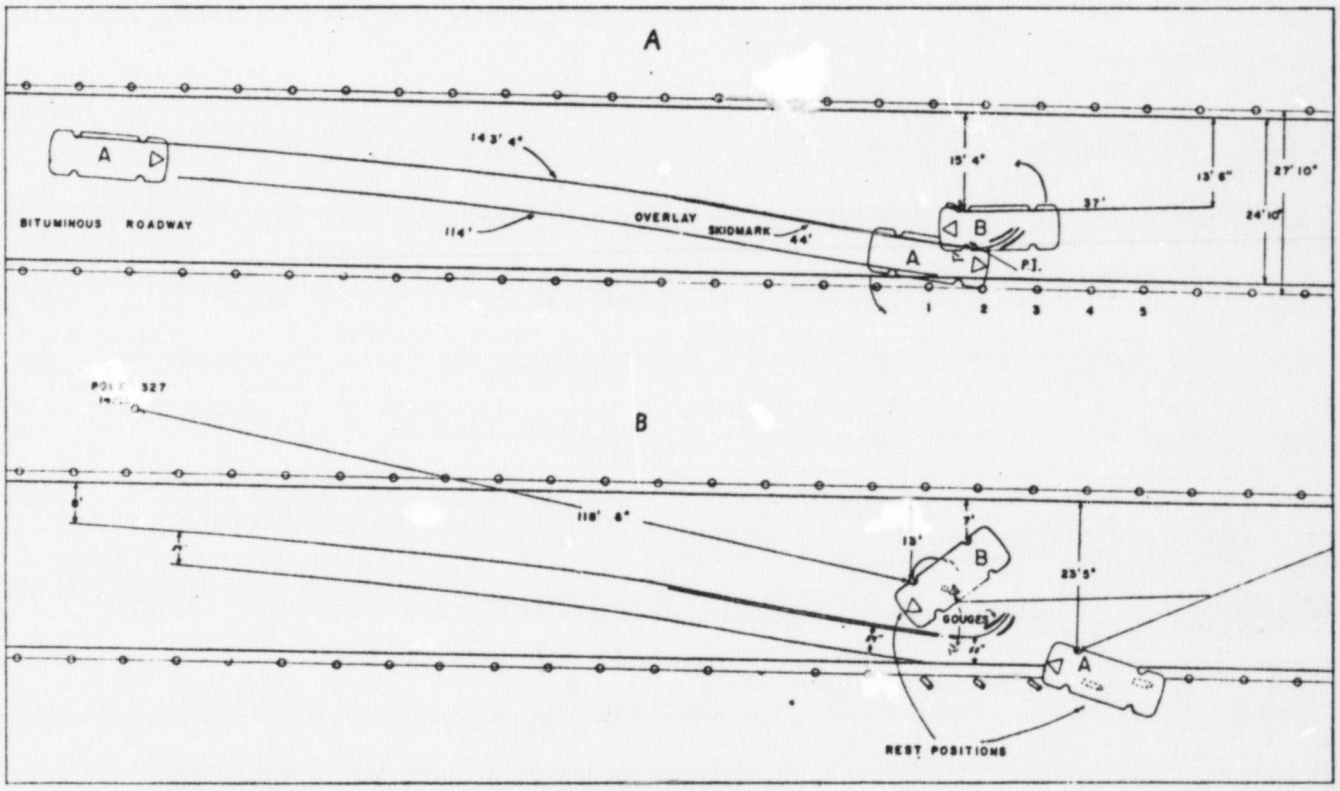
Vehicle deviation started with no anti-skid device and wheels locked.

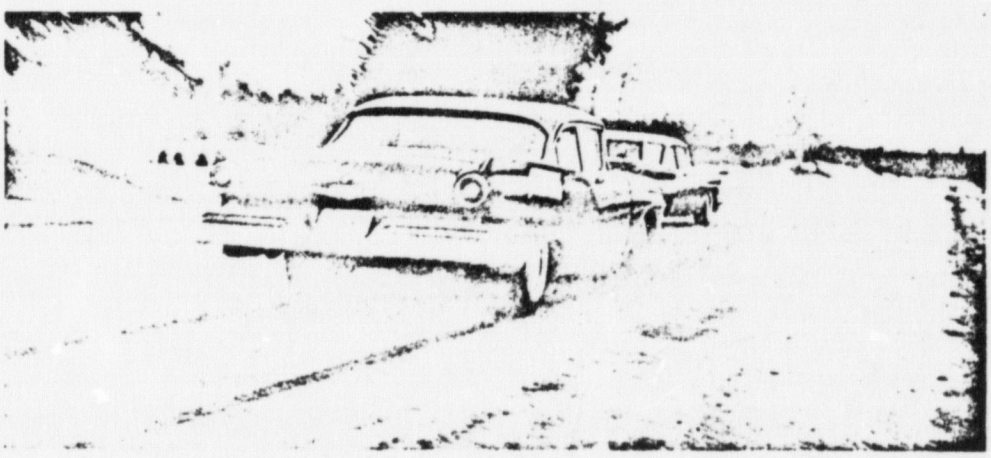
illustrated above are 6 tread designs each of which rely on a mechanical function of gearing the tread edges into the snow to obtain traction. When edges of cleats, blocks or chevrons become worn, the traction ability of the tire is reduced. The stopability of the tire is also reduced because the original sharp edges of the tread cleats become rounded and slide over the snow covered surface instead of interlocking.

Example of how Anti-Skid Braking System Could Have Prevented Collision That Resulted in 2 Deaths and Injury to 3 Persons. (MVR Files).

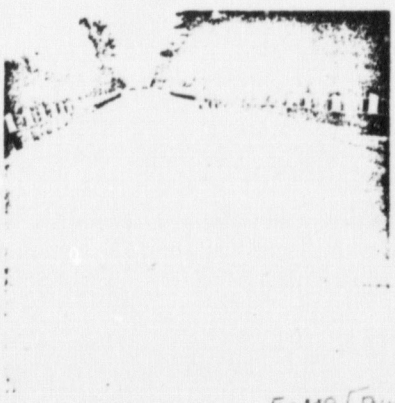
The drawings below show Car "A" (left on drawing) on the wrong side of the road traveling at relatively high speed as Car "B" (right on drawing) was on the wrong side of the road for some unexplained reason. The driver of Car "B" was observed to be crossing and recrossing the center line for some distance before impact. The driver of Car "A" concluded he should stop and applied the brakes fully causing the wheels to lock-up. The driver of Car "B" did the same thing about the same time. Car "A" skidded 143.4 ft. to a point of impact and Car "B" skidded 37 ft. to the point of impact. With the wheels of both cars locked they were both on a collision course and could not avoid impact by turning the steering wheel. The driver of Car "A" and a passenger in Car "B" were killed. Others received injuries but survived.

In this case, if Car "A" had been equipped with an anti-skid brake system he could have stopped in a shorter distance and also steered around Car "B" if he was not stopped before reaching Car "B". This is a prime example of the value of an anti-skid braking system.

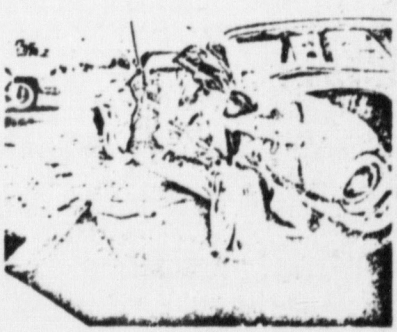




Emergency condition where car came out onto highway from left and stalled. Driver of other car tries to stop with locked-wheels and conventional braking system. With no steering control he will impact "stalled" vehicle.







LABORATORY ENDURANCE TEST

Anti-Skid Unit With Delrin Gear Operated Without Failure For 190,000 miles At 65 Miles Per Hour

A laboratory endurance type test was designed and conducted to evaluate the expected life span of the anti-skid unit operating over a large mileage period.

Units containing brass gears running against steel gears were tested together with Delrin gears running against steel gears.

The brass gear equipped units failed after operating for an average of 14,000 miles at 65 miles per hour while the Delrin gear equipped units were still running after 190,000 miles of operation.

The brass gear units became noisy after 5000 miles of use and this noise increased steadily up to the time of failure when a pronounced "howl" was heard. The Delrin gear units maintained a fairly constant noise level up to 190,000 miles.

The test bench consisted of a panel and platform on which 3 anti-skid units were mounted and coupled to an electric motor through pulleys, cables and gear boxes. The speed of the anti-skid unit was that which matched a vehicle speed of 65 m.p.h. The two Delrin gear equipped units were not changed or adjusted during the 190,000 mile test run. The brass gear equipped units failed during an average 14,000 mile test run and each was replaced by a new unit which then failed 14,000 miles later.

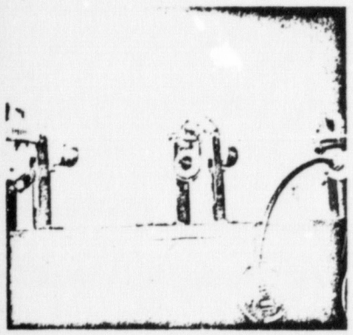
The two Delrin gear units were inspected after the 190,000 mile period and were found to be in what could be called "a good condition". The test bench motor was stopped and started every 8 hours. The motor was also stopped and started 6 times every 1000 miles to simulate intermittent vehicle use.

A total inspection of the Delrin units showed they were in condition to continue operating for many more miles.

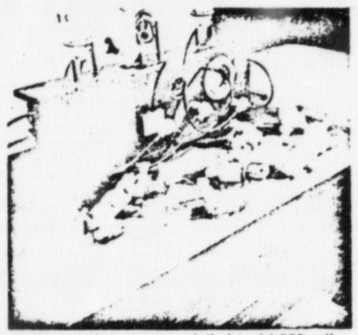
The test was designed to test a Delrin gear running against a steel gear but the balance of the unit was also involved.

The failures experienced by the brass gear units were confined to the gear teeth and the balance of the units was unaffected.

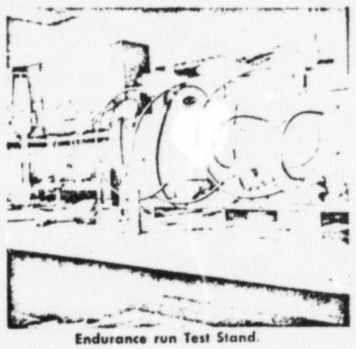
The test bench used is shown in the photographs in this section together with the assemblies used in the test.



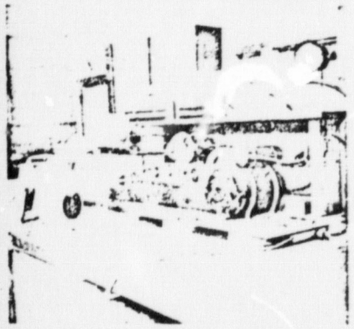
A Delrin gear ran 190,000 miles at 65 m.p.h.



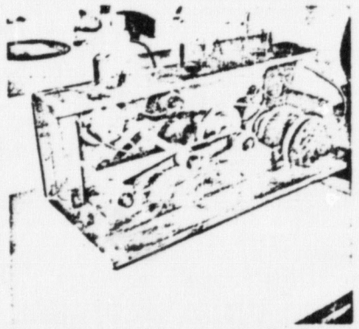
Units with brass gear failed in 14,000 miles.



Endurance run Test Stand.



Strobe Tach used to determine shaft speeds.

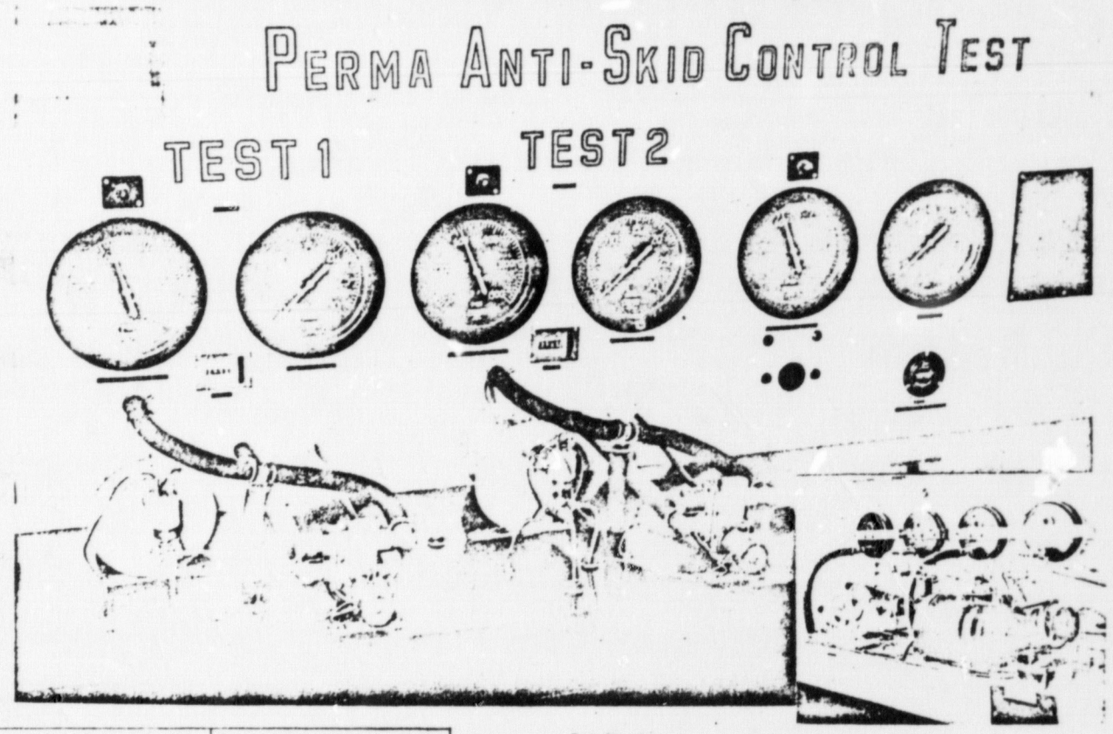


Rear of Endurance run Test Stand.



Brass gear unit that failed.

LABORATORY ACCELERATION-TORQUE TEST



Test Stand.

Rear of Test Stand.

A number of severe loadings and high torque factors are imposed on gear trains of a vehicle stopping from high speeds under emergency conditions. The governor assembly and related parts of the breaking system control become a gear train in the automobile system when this braking system control is installed. As such, it is subjected to severe accelerations and high torque values.

In order to properly test such a device under controlled conditions, a test bench was constructed that simulated conditions that would exist if a vehicle was accelerated from 0 m.p.h to 92 m.p.h and then decelerated from 92 m.p.h. to 0 m.p.h. in a time interval of 3 seconds. While such accelerations and speeds are not possible using the fastest land vehicle traveling on a roadway any product successfully surviving such a test would provide a high factor of safety. Such a test was made using a test bench shown in the photographs above.

Two complete Perma systems were positioned on a platform and connected to a hydraulic pressure system controlled by a hydraulic ram. The hydraulic pressures used ranged from 200 to 1000 p.s.i. to simulate pressures normally present when vehicle brakes are used. Pressure gauges labeled P.S.I. (located on the back panel of the test stand shown in photograph) indicated the pressures and changes in pressures during the test run.

A vacuum source to actuate the Perma power-cylinder was created by an 8 cylinder passenger car engine located outside the laboratory. Changes in vacuum was indicated by 8" dial vacuum gauges positioned on the back panel of the test stand.

A "D.C. Power Source" to [illegible] the extreme right end of the back panel.

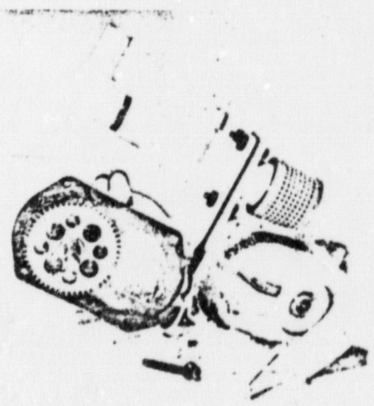
Electric cycle counters recorded the number of cycles during the test run.

The governor units were driven by cables connected to a shaft driven by a 110 volt A.C. motor. The speed of rotation of the governor units was recorded on a m.p.h. speedometer mounted on the back panel.

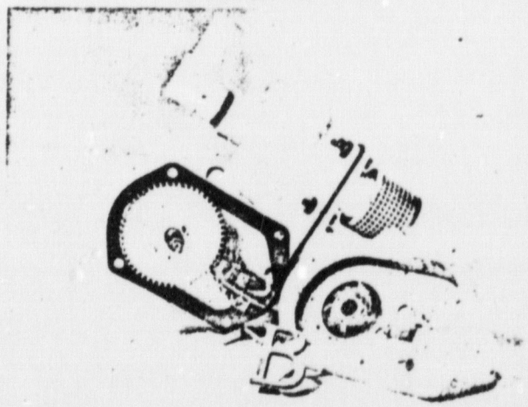
The electric drive motor ran at a constant speed of [illegible] r.p.m. An electric clutch, controlled by a timer switch, was imposed between the drive shaft of the electric motor and the shaft assembly driving the governor units. This electric clutch engaged and disengaged the electric motor shaft every 3 seconds causing the governor unit to accelerate to a rotation speed equal to a road speed of 92 m.p.h. and then down to 0 m.p.h.

In a series of tests, the noise factor of a brass gear versus a Delrin gear was studied. The brass gears started to become noisy after 1200 to 1500 cycles and the noise level increased steadily up to 30,000 cycles. The Delrin gear noise level remained fairly constant over the cycle range.

Illustrated here is a typical example of the condition of an assembly with a brass gear and another assembly with a Delrin gear after 30,000 cycles from 0 m.p.h. to 92 m.p.h. to 0 m.p.h. in a 3 second period per cycle. The cleanliness of the housing interior containing the Delrin gear can be seen in the photograph shown here and the heavy accumulation of brass dust can be seen in another photograph showing another assembly that contained a brass gear. Both gears ran against a steel gear. The ability of the Delrin gear to function for 30,000 cycles with practically no wear was the basis for its



Brass gear unit that failed.



NO MATERIAL ON PAGE E-151

PLAINTIFF'S EXHIBIT 20

AUTOMOTIVE NEWS, JANUARY 28, 1963

Engineering Production Materials

Antiskid Device for Autos Developed by Eastern Firm

AUTO makers and suppliers have been trying to develop an effective antiskid device for cars for years, but now a small Massachusetts research firm has begun making such a mechanism. The company calls it a "major safety breakthrough."

Developed by Perma Research and Development Co., North Attleboro, Mass., this antiskid unit basically operates like the other auto devices under development and like the antiskid systems used so widely on aircraft.

When the vehicle's brakes are applied, the brake shoes press against the brake drums almost until the wheels stop revolving. One component of the antiskid unit then senses this impending skid, and another component automatically releases the brakes slightly, preventing the skid.

• • •

FRANK PERRINO, president of Perma Research and inventor of the device, said the elimination of skidding does two things. It insures that the driver will have steering control of his vehicle throughout the stop. And, more importantly, it reduces stopping distances—in some situations.

According to Perrino, the antiskid unit achieves its most dramatic results on snow or other slippery surfaces, with reduction of stopping distances up to 50 percent. Somewhat shorter stopping distance reductions are likely on ice or during rain.

Outstanding stopping results were achieved recently at a demonstration at a race track near Wilmington, Del., but this was on a road made extremely slippery with soap and water.

However, on dry concrete or asphalt—by far the most common motoring situation—Perrino said that in about half the cases stopping distances are not reduced. The stopping distances actually would be extended in some cases.

Perrino said that size, weight and cost considerations have been the principal deterrents to the adaptation of such an antiskid device for automotive use.

• • •

ANOTHER major deterrent, especially in the case of an antiskid device under development by Bendix Products Division of Bendix Aviation Corp., is that the engineers were not satisfied with any device that extended the stopping distances on dry surfaces, even when it did reduce distances on slippery surfaces. Apparently this compromise was acceptable to Perrino.

Still, this device is probably worthwhile and it is one that took considerable engineering ingenuity. For example, the "brain" in the unit not only has to sense an impending skid, it also must determine the kind of road on which the vehicle is travelling.

Behind this is the fact that much more brake pressure (about 700 pounds at 50 miles an hour) is needed to stop the average vehicle on a dry concrete road than is required on an icy road, where about 200 pounds of pressure might be adequate at the same speed.

To get the best nonskid braking, the device has to "know enough" to reduce the brake pressure to about 675 pounds on a dry surface and to about 190 pounds on a slippery surface.

• • •

THIS antiskid unit, which is covered by several patent applications, basically accomplishes its mission with a gyro governor driven by a cable hooked to the transmission, much as a speedometer is.

As a result, the spinning of the gyro precisely reflects the spinning of the wheels, although the gyro does spin seven times faster than the wheels.

When the wheels stop turning, the gyro immediately stops spinning and collapses. In a fraction of a second, this causes the device's other major component—the control valve—to reduce the brake pressure slightly and make the brakes release just before the wheels lock. The wheels then begin rolling again and this type of brake-pumping cycle continues until the vehicle stops.

The control valve does its job by interrupting the hydraulic line between the master cylinder and the wheels. The third major component is an electrical solenoid which is a fail-safe device providing braking if the antiskid unit becomes inoperative.

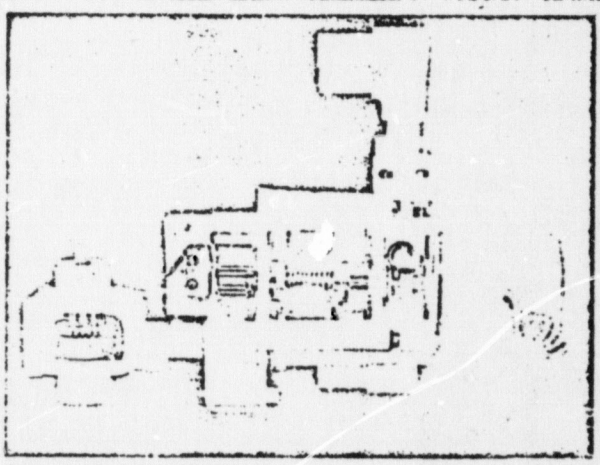
Perma's antiskid unit weighs less than two pounds and can be held in the palm of a hand. Most of its parts are made of Delrin acetal resin and Zytel nylon.

Delrin acetal resin was used for 11 parts because of its bearing properties, good strength, lightness and resistance to hydrocarbons, while Zytel was used for four parts because of its toughness, resilience and abrasion resistance.

Because of the selection of these materials, duPont Co. engineers provided development and testing assistance.

Perma antiskid units are in production for late-model cars. Installation requires about two hours. Perrino is lining up dealerships and expects to have 1,000 outlets by the end of January.

Installed cost of the device is $80 to fleet owners and $119 to $129 to individuals, including the cost of a Bendix Hydrovac power-brake unit. Perrino said prices for cars with power brakes could be reduced by about 50 percent.



Perma's Anti-Skid Unit—

The "brain" of Perma's anti-skid device is this gyro governor which spins whenever the wheels spin. When the gyro stops spinning, a control valve regulates the brake-line pressure to give maximum braking short of wheel-lock.

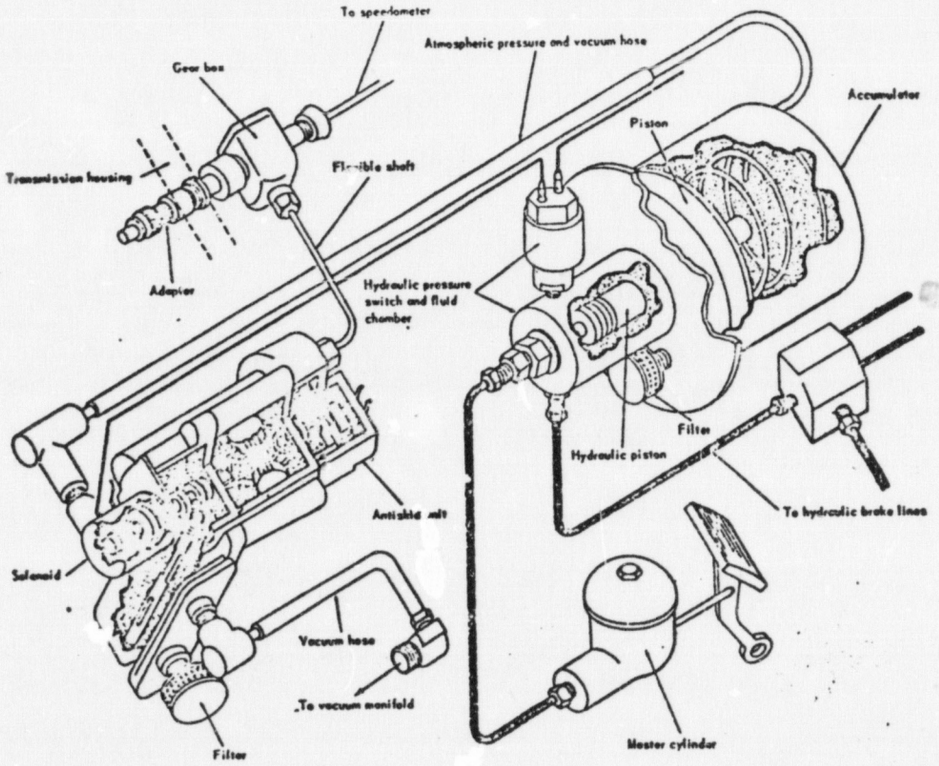
A simplified antiskid device uses engine vacuum to control a variable accumulator for release of braking pressure in event of wheel lock-up. The redesigned unit, quickly installed on cars with or without power brakes, is a self-contained accessory that modulates brake-line pressure to a point just below the skid-producing pressure.

The original unit (DESIGN NEWS, April 11, 1960) provided combined automatic braking and skid control through the accelerater pedal. A later design was built around a power brake. Engine vacuum in both cases modulated brake pressure as regulated by a rotary valve. The latter effected release of skid pressure through a governor that responded to wheel lock-up. A safety valve allowed brake fluid to bypass the power brake and provide adequate pressure in event of engine failure or stall. In addition, damping was provided for the driver who wanted to avoid the sensitive feeling of the power brakes.

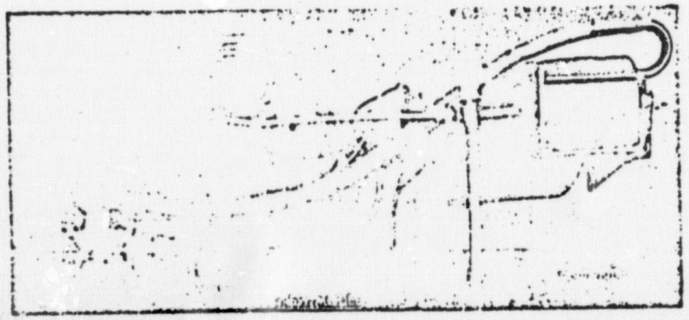
The redesign is independent of a specific braking arrangement and functions as an easily installed accessory. The antiskid unit is the same, except that now it simply regulates a separate variable accumulator, only in event of a skid condition. Engine vacuum releases the brake-line pressure to just below the skid level when a wheel lock-up condition occurs. Vacuum is fed to the release side of the accumulator through the rotation of the rotary valve in the antiskid unit. At the same time, closing of a valve in the hydraulic section of the accumulator prevents overriding of the antiskid system by the driver.

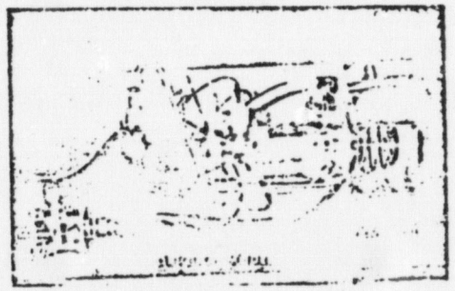
There is no change in the "feel" of the brake pedal after the unit is installed. Also, since the device does not rely on vacuum for braking power, there is no need for the safety valve and associated lines and fittings.

The redesigned antiskid control is a product of Perma Research and Development Co., North Attleboro, Mass.



SELF-CONTAINED UNIT *supplements existing braking system. Governor senses locked wheel condition and modulates rotary valve. Solenoid assures that antiskid system will be operative only during skid.*





UNIT WITH POWER BRAKE *required replacement of existing braking system and complicated installation.*

Reprinted from the October 16, 1963 issue of DESIGN NEWS, published by Cahners Publishing Company, Inc.

E-153
[PX 20, p 21]
[PX 20, p 21]

A new safety device made of "Delrin" and "Zytel" promises straight, controlled stopping even when a driver brakes on icy roads

Sudden Stops Without Skidding



Reprinted from
DU PONT MAGAZINE

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Better Things for Better Living
. . . through Chemistry

Sudden Stops Without Skidding

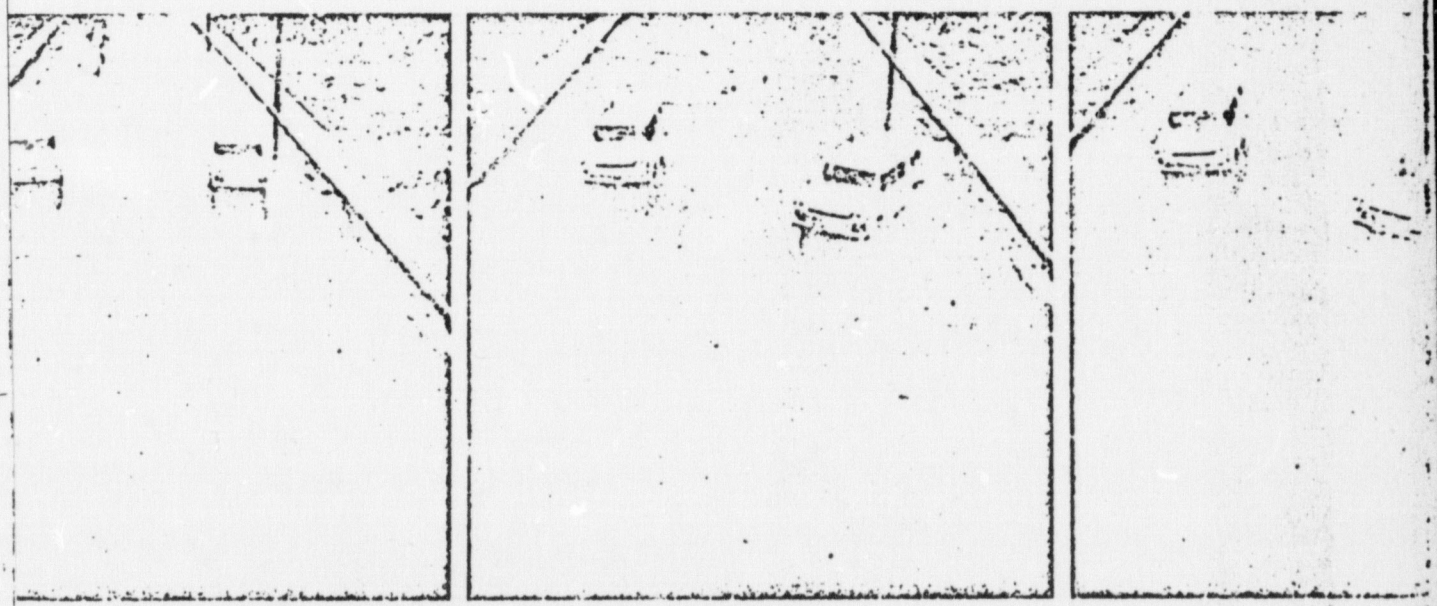
A new safety device made of "Delrin" and "Zytel" promises straight, controlled stopping even when a driver brakes on icy roads

One day last winter, two station wagons of the same make and model sped side by side down a New England road whose wet surface was rapidly becoming a ribbon of ice. Onlookers tensed as, at a predetermined signal, both drivers slammed on their brakes.

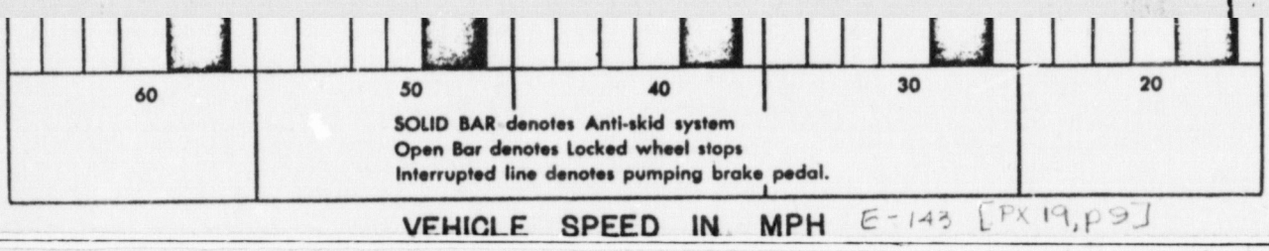
As the film strip below recorded, one vehicle swerved immediately, spun sideways, then slid out of control across the center line and into the other traffic lane. But there was no collision, for the other station wagon had come to a quick, unwavering stop a safe distance away.

The startling difference in the behavior of the two vehicles was attributable to a new automotive safety device that promises to end skidding when drivers try to stop on slippery roads or at high speeds on wet or dry pavements. Dubbed the Perma Anti-Skid Unit, the small device compensates for a standard human reaction: applying the brakes so hard that wheels lock and slide like sled runners.

The new Perma Anti-Skid Unit is triggered by its "brain," a gyro governor that spins with the car's turning tires. When the tires lock, the gyro instantly stops. This ac-



In the sequence of photos above, similar test vehicles traveling side by side make sudden stops on a slippery highway. Car, left, contains a new automotive anti-skid device that brings it to a safe, straight stop. Wheels on the standard vehicle, right, lock, causing it to skid.



E-143 [PX 19, p 9]

[PX 20, p 5]

ivates a control valve that regulates brake pressure between the master cylinder and th- wheels. The valve alternately tightens and releases the brakes, producing a pumping action that prevents wheel lock while providing maximum deceleration and steering control.

"A driver gets most benefit from his brakes when he can hold his tires just on the verge of locking," explains the unit's inventor, Frank Perrino, president of Perma Research and Development Co., North Attleboro, Mass. "But even if he keeps his wits in an emergency, the average driver doesn't have the time or skill to analyze the road surface and apply the precise pedal pressure required for a safe, effective stop. With the anti-skid unit, however, he can slam on the brakes and let the unit do the thinking for him.

"For e[illegible]mple, the wheels of an average car travel[illegible]g 50 mph on dry pavement will [illegible]ck with about 700 pounds of brake pressure. About 200 pounds of pressure will lock the wheels at the same speed on ice. The new unit prevents skidding by modulating pressure around 675 pounds on the dry surface and 195 pounds on the icy road, no matter how hard the driver pushes on the foot pedal.

"Moreover, the unit is sensitive enough to adjust to rapidly changing road conditions. If ice or snow suddenly gives way to dry pavement, the gyro governor senses the change and reapplies additional brake pressure to take advantage of the improved traction. Under normal driving conditions, the unit has no effect on brake operation."

In the past year, Perrino has successfully demonstrated the Perma Anti-Skid Unit hundreds of times on road surfaces ranging from dry pavement and gravel to wet asphalt and ice. This winter, the device is being tested for certification by safety councils, insurance companies and independent automotive testing agencies. Perma is producing units for late model cars and is establishing a marketing network.

Perrino set out to end auto skidding after a harrowing experience while he was an Air Force mechanic overseas. He was driving down a wet German highway when a child suddenly darted in front of his car. Reacting normally, Perrino slammed on his brakes, went into a long, uncontrolled slide and scarcely missed hitting the child.

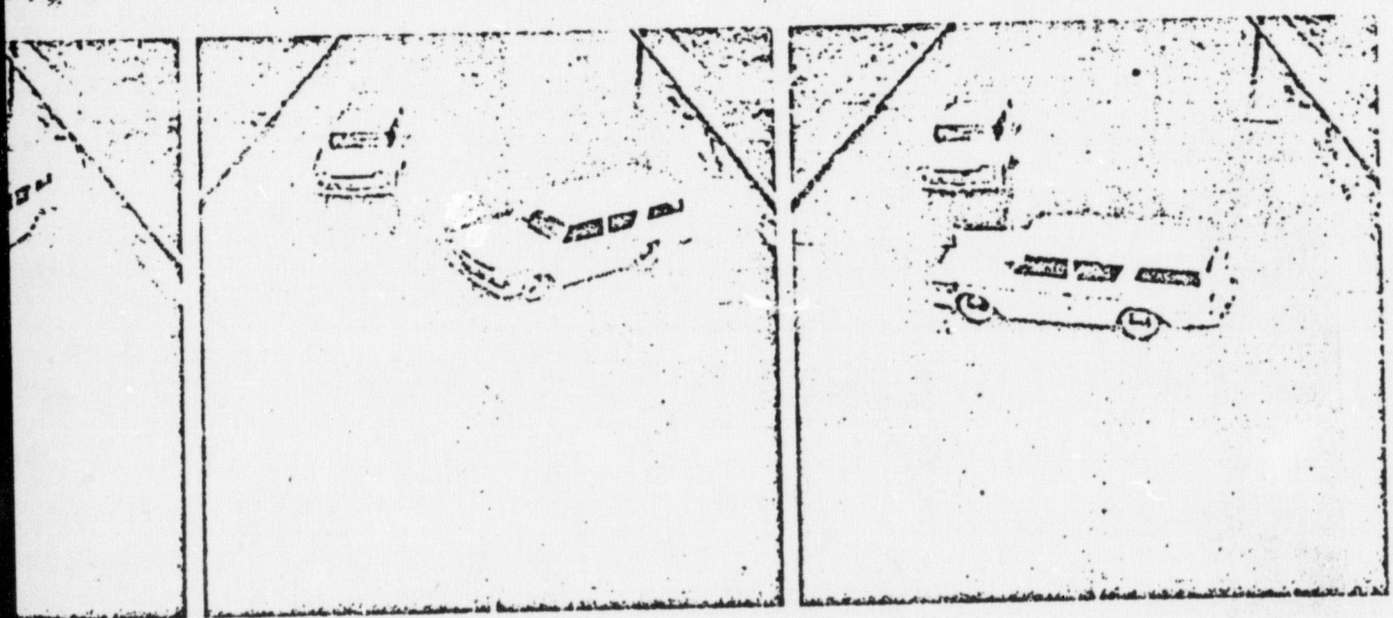
Familiar with the anti-skid systems used to prevent aircraft landing wheels from locking, Perrino opined that a similar device should be available for automobiles. He found, however, that size and weight alone ruled out adapting the aircraft system to cars.

"Besides being bulky, anti-skid systems on aircraft cost more than most cars," he explains. "To be acceptable, an automotive unit would have to fit comfortably under the hood and sell in the same price range as standard accessories."

After more than five years' work, much of it part time, Perrino developed the present Perma Anti-Skid Unit. Attaching both to power and conventional braking systems in standard and compact cars, it weighs less than two pounds and costs no more than a good radio or heater. The unit should cost less, he says, when it's in full production.

The young inventor overcame many of his early design and manufacturing problems by utilizing unit housings and many working parts molded of "Delrin" acetal resin and "Zytel" nylon resin. Components are molded by Nylon Products Corp., Clinton, Mass., and Danielson Manufacturing

continued





Due to differences in surface slipperiness under each wheel car starts out of control with a conventional braking system.

Car rotation is advanced to a point where control is no longer available and car will continue to rotate.

E-144

[PX 19, p10]

Co., Danielson, Conn.

"Our first unit was cast in metal," Perrino recalls, "but it proved to be heavy and expensive. We ran into high costs for machining, tooling and assembly. We also had the problem of lubricating bearings encased in a sealed unit.

"By redesigning in 'Delrin' and 'Zytel', we found we could cut costs 40 per cent and get the necessary strength with half the weight. At first, a few intricate parts gave us some molding problems, but assistance from Nylon Products and Du Pont ironed out those kinks.

"Now all machining is eliminated. All threads are precision molded, and self-tapping screws can be used for fasteners in many places. We don't need separate bearings because 'Delrin' and 'Zytel' have low coefficients of friction and don't have to be lubricated. Moreover, neither 'Zytel' nor 'Delrin' is affected by petroleum hydrocarbons or heat from the engine.

"Besides money and weight advantages, our new unit has improved durability and performance. In comparative tests, brass gears in our earlier model failed after some 40,000 simulated emergency stops. Our new plastic gears have been tested 70,000 times without any sign of weakening."

Plastic Pressure Switch

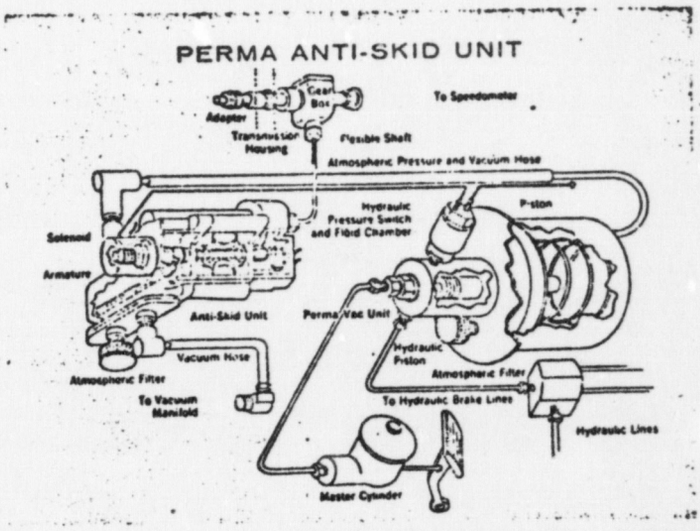
Indicative of the ingenuity applied to the Perma Anti-Skid Unit is a new pressure switch that converts pressure readings to electrical signal controls. Perrino found that the only adequate commercial switch cost $30 and weighed nearly three pounds, a pound more than the skid-control unit itself. He then designed a new pressure switch of "Delrin" and "Zytel" that costs a fraction of its metal counterpart and weighs only 3.5 ounces. Because the switch operates at any range from 0 to 3000 psi, it's also being used on hydraulic starters for diesel engines, in aircraft, and in industrial applications where electrical circuits must be closed at predetermined pressures.

Since perfecting his anti-skid device, Perrino has been demonstrating its merits across the country to safety engineers, state officials, government agencies and automobile manufacturers. Leaving no stone unturned, he has submitted his invention to automotive testing experts, who are running units on the road and in the laboratory as well.

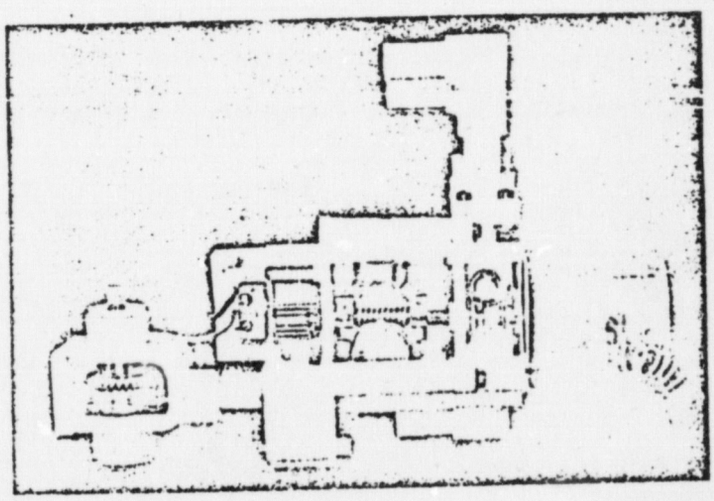
One interesting battery of tests, according to Perrino, gives a unit more punishment in one hour than it would normally receive in an operating lifetime.

"Units mounted on test stands are constantly accelerated from 0 to 80 mph," he says. "Then the brakes are slammed on to simulate a driver making an emergency stop at high speeds. Other units are taken directly from freezers and raced to 80 mph before they can warm up. Still other units are operating around the clock in temperatures ranging from −30° to 275°F., simulating long-distance driving in all climates."

To date, road and laboratory tests from all quarters are bringing excellent results. Concludes Perrino, grinning, "You might say it's a case of a business speeding up by making sudden stops—without skidding." ■

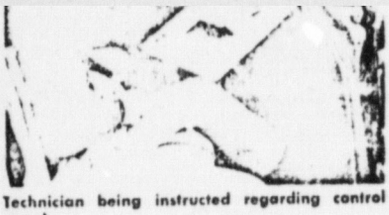


A car's wheels lock when a driver puts too much pressure on the brake pedal, bottom. The gyro governor in the anti-skid unit, left, senses such locking through a connection to the transmission, top. By triggering its solenoid and armature, it sets off a valving action at the Perma Vac unit, right. This modulates fluid pressure in the brake system, thus creating a pumping action.

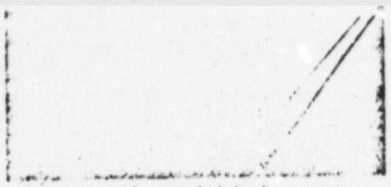


Device's gyro governor "brain" is protected by a housing of "Delrin".

Copyright 1963 by E. I. du Pont de Nemours and Company, Wilmington, Delaware 19898

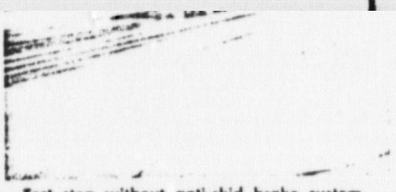


Technician being instructed regarding control panel.



Test stop with anti-skid brake system.

E-145



Test stop without anti-skid brake system.

[PX 19, p 11]

[PX 20, p 7]

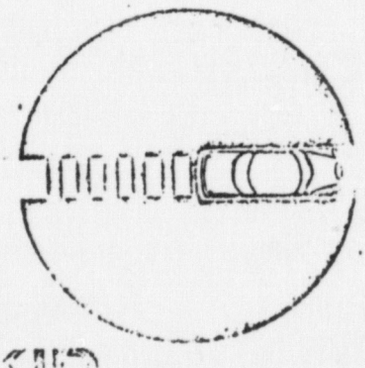


NO SLIP

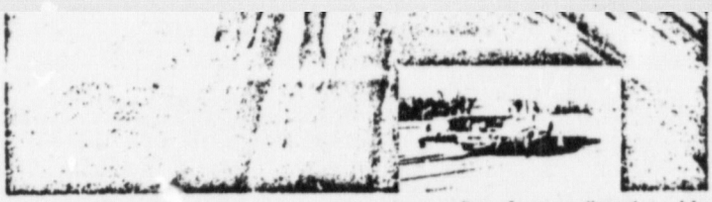
NO SLIDE

NO SKID

UE TO OVER-BRAKING



RAIGHT SMOOTH CONTROLLED STOPPING with . . . PERMA ANTI-SKID CONTROL



E-142 [PX 19, P12]

Photograph shows the impending skid condition that exists when anti-skid unit is used. Interruptions in solid tire marks shows cycling of system. (insert) precision steering was possible while emergency braking was present.

PERMA ANTI-SKID CONTROL



PRINCIPLE OF OPERATION

The master cylinder is the reservoir from which fluid is delivered whenever the brake pedal is depressed. The anti-skid unit is the device which prevents an excess of pressure from reaching the wheels, which would cause a skid.

The anti-skid unit employs a gyro governor that always rotates in direct relationship to the vehicle's wheels and speed of the vehicle.

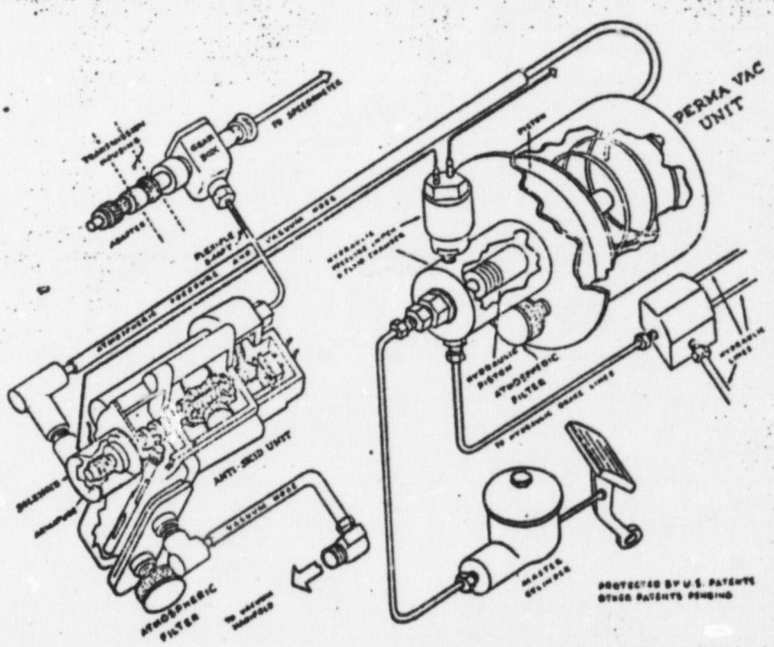
The governor does two jobs; it senses a locking wheel condition by the loss of its own rotation in relationship to vehicle speed and transfers this message to the electrical solenoid and armature to allow the unit to instantly go into operation.

Secondly, it controls the operation of the Perma-Vac through its valve mechanism. This valve under control of governor position modulates fluid pressures.

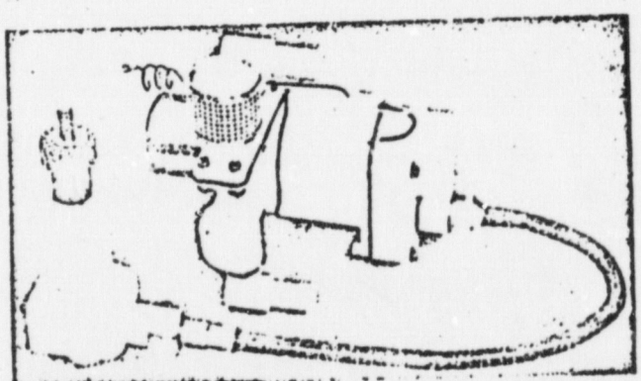
EXAMPLE: To start the cycle of the anti-skid in operation, we will assume a vehicle speed of 50 MPH, bearing in mind that the more slippery the road surface, the less pressure is required to lock the wheels, which causes the skid. On a dry road 700 P S I of fluid pressure would lock the wheels; the governor must relieve just enough pressure to allow the wheels to turn, yet stop in the shortest possible distance utilizing all road-to-tire friction available. Thus, the 700 P S I would be reduced to 650 to accomplish this end. If during this type of dry pavement stop employing 650 P S I, the vehicle approached an icy patch, the governor would automatically reduce the brake pressure to compensate for the more slippery surface.

The unit also employs fail-safe features for added driver protection.

SCHEMATIC DIAGRAM of
PERMA ANTI-SKID CONTROL



Schematic shows a standard master cylinder, Perma Vac unit, anti-skid unit, and anti-skid drive assembly. (adapter and gear box)



Featuring a gyro governor, it connects directly into brake system and transmission.

"MECHANICAL BRAIN"

The "Mechanical Brain" is a small, but complex mechanism featuring a gyro governor, which is connected directly into the brake system and transmission. The unit is able to sense in a split second the decrease in wheel revolutions that accompanies a "wheel lock" condition when brakes are jammed on a slippery surface. The anti-skid unit reacts immediately to sufficiently relieve brake pressure while imparting the pumping action which allows the driver to retain steering control during deceleration.

Vehicle deviation started with no anti-skid device and wheels locked.

E-147 [PX19, p13]

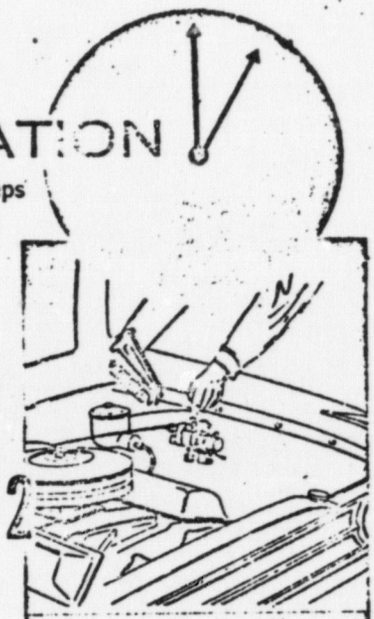
[PX 20, p 9]

ONE HOUR INSTALLATION

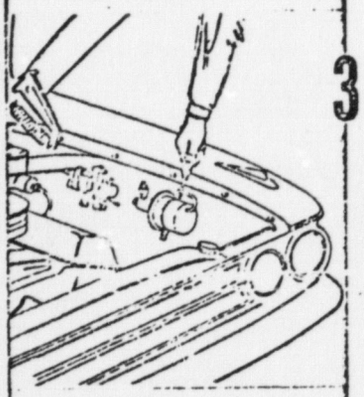
For Most Cars — in 8 Simple Steps



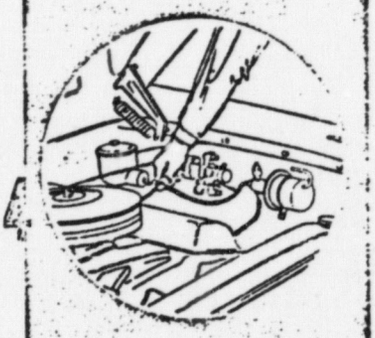
INSTALL GEAR BOX Remove speedometer drive cable and adapter from transmission. Replace with "Perma" gear box and adapter. Connect speedometer and Anti-Skid control drive cable to gear box. Feed Anti-Skid cable from gear box to engine compartment by fender wall.



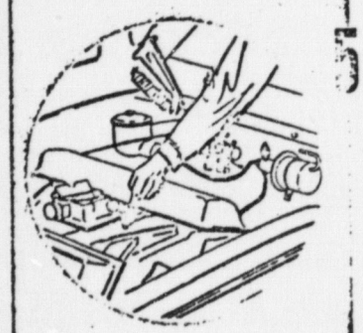
INSTALL "MECHANICAL BRAIN" Attach "Mechanical Brain" to cable. With slight slack, attach "Mechanical Brain" to fender wall, with sheet metal screw and lock washers. Make sure there are no sharp bends or kinks in the cable.



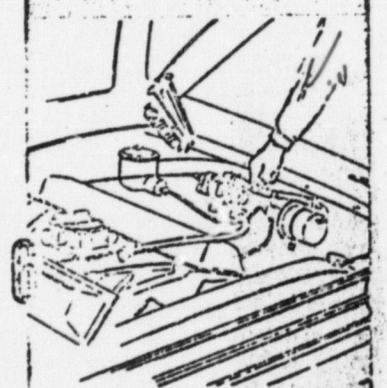
INSTALL "PERMA-VAC" Place template in specified position. After drilling holes indicated on template, mount "Perma-Vac" in position with screws supplied.



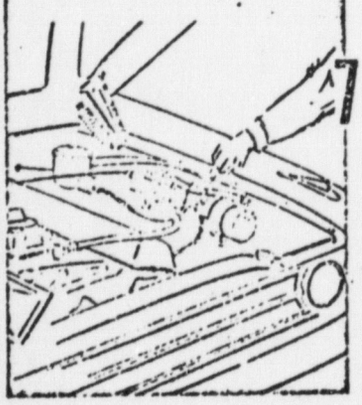
HYDRAULIC LINE CONNECTIONS Remove hydraulic line from front of master cylinder and attach hydraulic line from the front of valve body on "Perma-Vac" to the master cylinder. Attach line from side of valve body on "Perma-Vac" to the wheel cylinder connection block. Bleed system at "Perma-Vac" valve body connections as you would in any conventional vehicle, then refill master cylinder with brake fluid.



INSTALL VACUUM MANIFOLD FITTING Remove carburetor air filter in order to expose intake manifold, install supplied fitting into manifold. If no threaded hole is provided in manifold, drill hole and tap in any centrally accessible location.



INSTALL VACUUM HOSES (2) Remove color coded hoses from kit. Connect Red hose to the fitting on the intake manifold and to the elbow on the same side as the filter on the "Mechanical Brain." Connect the White hose from the other elbow on the "Mechanical Brain" to the U-Bend tube at the rear of the "Perma-Vac." (Use hose clamps on all connections)



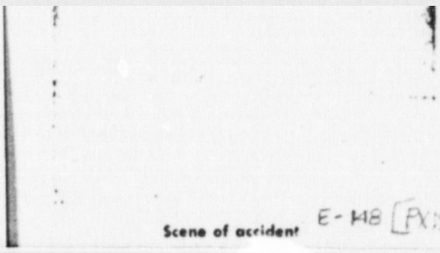
ELECTRICAL CONNECTIONS Locate 12 volt source that completes circuit only when Key is "ON." Connect this source to hydraulic pressure switch with wire connections provided. Connect wire from other terminal of hydraulic switch to wire protruding from "Mechanical Brain" solenoid. (TAPE ALL CONNECTIONS)



ROAD TEST Produce skid situation by braking the vehicle suddenly and locking the wheels, on loose gravel or slippery pavement. You will feel the pulsation of the brakes releasing when the wheels lock.

CHECK ALL FITTINGS AND CONNECTIONS

Your customer can now drive away with the confidence his vehicle brakes are controlled by the revolutionary Perma Anti-Skid unit.



Scene of accident E-148 [PX 19, p 14]

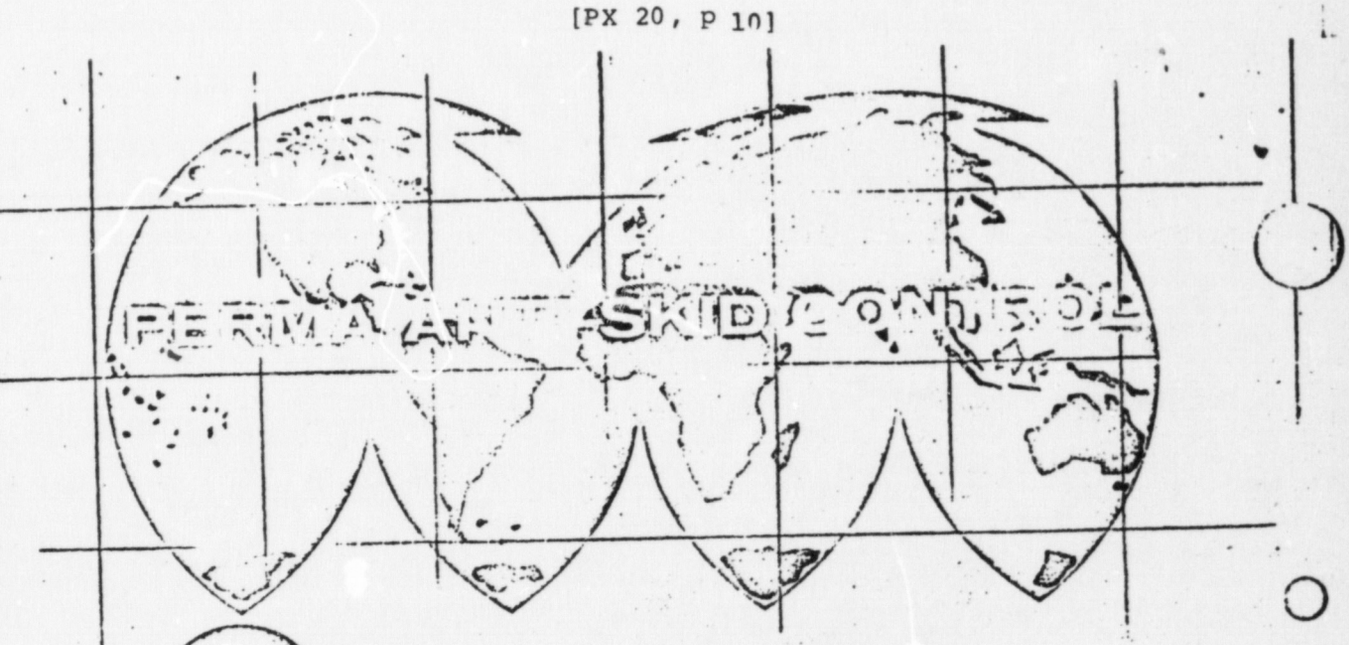


Impact to Car B (front to rear)



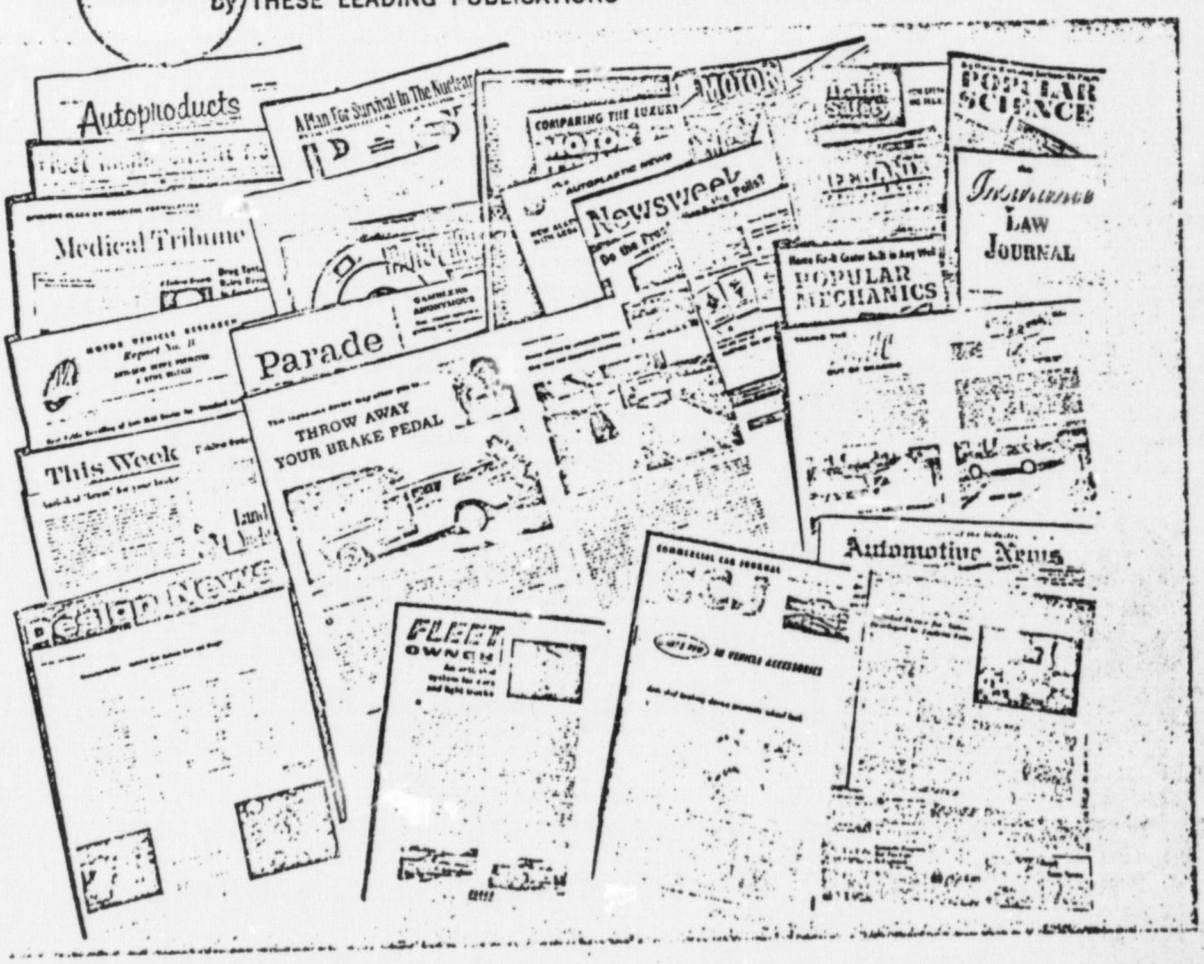
Impact to Car B (rear to front)

[PX 20, P 10]



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RESEARCH AND DEVELOPMENT COMPANY

"Advanced Control Systems"

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LITHO. IN U.S.A.

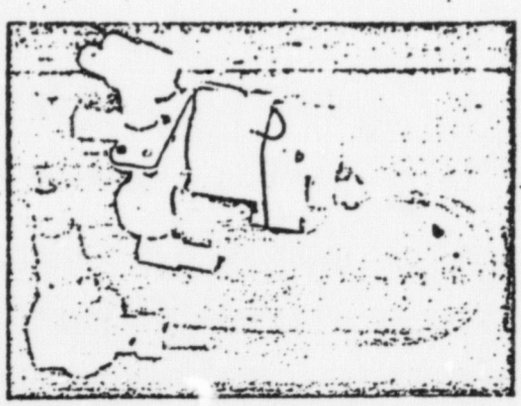


E149 [PX 19, p 15]

[PX 20, p 11]

FLEET OWNER

An anti-skid system for cars and light trucks



Featuring a gyro governor, it hooks directly into brake system and transmission.

Have you ever had reason to brake suddenly on a slippery pavement to avoid a collision? If so, then you might have had the experience of sliding sideways in spite of all efforts to keep the vehicle straight. Still worse is a complete spin and then a skid out of control.

An anti-skid system was tested under these conditions by FLEET OWNER's chief editor, Carroll Boyce, last month at Delaware Park race track near Wilmington. To simulate a slick pavement DuPont—sponsor of the test for Perma Research and Development Co.—"lubricated" the test area with a mixture of soap and water.

With the anti-skid controls inoperative, sudden breaking by Boyce produced the skidding action shown below in the photo at left. Jamming the brake at the same speed but with anti-skid controls "on," Boyce was able to bring the test car to a controlled straight stop (photo at right). A skid was prevented because the test car responded to the "mechanical brain" of the anti-skid unit.

The sensing unit or "brain," so to speak, consists of a tee adaptor for the transmission's speedometer drive opening and a gyro governor hooked directly into the brake system. Both units are connected through their own cable drive. Application of the anti-skid system is limited to vehicles with hydraulic or air-over-hydraulic brake systems.

The speedometer adaptor, acting as a power take-off, spins the gyro governor at ultra-high speeds. When brakes are jammed on a slick surface and the wheels go into a "locked" condition, the governor senses the drop in wheel revolutions in a split second. The anti-skid system immediately takes over to sufficiently relieve the high brake pressure which locked the wheels by delivering a pumping action on the brake system. The pumping or pulsing action is at the rate of 20-milli-seconds. This "on" and "off" cycle of brake pressure relief allows the driver to retain steering control during quick deceleration.

Engineers have worked on the skidding problem for years but it still remains high as a cause of highway fatalities. As inventor of the unit and president of Perma Research, Frank Perrino puts it, "No one has yet succeeded in devising a way to speed up a driver's reaction time."

A system similar in principle has been used on commercial aircraft for some time to solve braking problems. The cost for a modern jet system is more than the cost of most automobiles. For automotive application, the price range is around the cost of a car radio or heater accessories.

A major factor has been DuPont's engineering assistance in cost reduction and product reliability. DuPont's "Delrin" acetal resin and "Zytell" nylon resin are featured in many segments of the anti-skid units. Delrin chosen for bearing properties, lightness, strength and resistance to hydrocarbons. Zytell used in the gears was chosen for its toughness, resilience and abrasion resistance.

SKIDDING in test car on this soap and water slicked road is FLEET OWNER's chief editor Boyce. He's just jammed on the brakes at 35 miles per hour.



NON-SKIDDING this time, same road. He's driving test car equipped with anti-skid device which lets him come to controlled stop.



FLEET OWNER

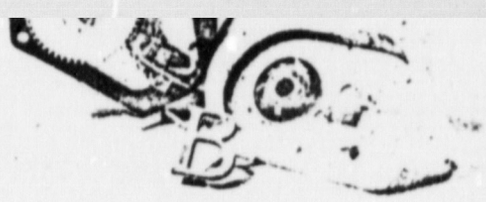
Reprinted from

sures during the test run.

A vacuum source to actuate the Perma power-cylinder was created by an 8 cylinder passenger car engine located outside the laboratory. Changes in vacuum was indicated by 8" dial vacuum gauges positioned on the back panel of the test stand.

A "D.C. Power Source" to energize the solonoids can be seen in the small photograph above. The unit is shown at

Delrin gear can be seen in the photograph shown here and the heavy accumulation of brass dust can be seen in another photograph showing another assembly that contained a brass gear. Both gears ran against a steel gear. The ability of the Delrin gear to function for 30,000 cycles with practically no wear was the basis for its being selected as a material for this unit.



E-150

A. Delrin gear unit

[PX, M, p 16]

[PX 20, p 12]

ASSOCIATION OF CASUALTY AND SURETY COMPANIES

MOTOR VEHICLE BULLETIN

110 WILLIAM STREET • NEW YORK 38, N. Y.

Section	Bulletin Number	Replaces Bulletin Number
7.0 Safety Devices and Accessories	7.2--11	

7.2--11: Perma Anti-Skid Control device for hydraulic brake systems

The device described in the attached literature prevents locked-wheel skids and is applicable to the hydraulic brake systems of passenger cars. Systems similar in principle have been used for some time in aircraft applications but have hitherto been uneconomical as to size, weight and cost for use on small road vehicles.

The two basic advantages of such a device are as follows:

1. It makes possible a braking effort which approaches the practicable maximum under given road surface conditions. Maximum brake effort is theoretically attained at the point referred to as impending skid. There is no practical way for a driver to know when this point is reached, so most drivers in practice either go into a locked-wheel skid or settle for a brake effort considerably short of what is theoretically available.

   The effect of the Perma device is to produce, under maximum brake application, a repeated skid/release cycle, thereby gaining the greatest possible braking effort; this is accomplished, further, with no need for the driver to analyze road surface conditions or other factors which might be relevant to a brake application.

2. Locked front wheels have no steering value; locked rear wheels track badly, tending to break away out of control.

   The Perma device prevents locked skids and therefore makes possible continued steering control with the front wheels and assures proper tracking by the rear wheels.

In general, then, the advantages of such a device include achievement of maximum feasible braking action on any type of road surface under full brake application and the maintenance of vehicle control during such braking. Details of the device are included in the attachments; further information can be obtained from the manufacturer.

[PX 20, p 12]

CADILLAC ENGINEERING -'63 -'64 Doug Wing
Dave Lawson

INC-7163
WO 5954-5

- Perma Anti-Skid Tests

[PX 21, p 1]

Stopping Distances -- Comparison, Perma Anti-Skid.

Road Condition	Speed mph	Stop	No Anti Skid	Anti Skid On R only	Anti Skid On All 4 power
Dry - High μ	40	panic	75-80		81-78
" "	"	panic	71-75	70-74	
" "	"	control'd	76-81		
Wet - High μ	"	panic	86-91		74-77
Gravel - dry	"	panic	90-91		96-105
Wet - Low μ	"	panic	118-132		98-114
Snow on Gravel	"	panic	132-206		148-175
4" Snow on Paved	35	panic	212-240		198-217

Note: stopping distances are min and max obtained in 3-5 tests at each condition. All stopping distances in feet, recorded with fifth wheel.

Plaster
2 Rd.
4% Rd.
8% Rd.

Desired Press. R. 1100 - Max

Bob Van Horne (FR yist)
Delco Moraine

Doug Wing
Fred Yeitt Delco
Dave Lawson
~~Fred Hillier~~

[PX 21, p 2]

E-164

PLAINTIFF'S EXHIBIT 21

PLAINTIFF'S EXHIBIT 22

Dated March 13, 1964

PERMA RESEARCH & DEVELOPMENT CO.

"ANTI SKID" - CONTROL

P-1881-RK

RECEIVED
MAR 16
MACHINE
ENGINEERING DEPT.

Attended: Mr. C. E. Brocker
Mr. S. F. Barbara
Mr. P. Dulk
Mr. C. R. Fricke
Mr. J. Gomes
Mr. R. A. Kloby
Mr. T. F. Lindmar
Mr. J. P. Moore
Mr. G. Plungis
Mr. A. Romel
Mr. R. Soltys
Mr. E. Sprague

Summary

If this unit is produced at Singer, we will be the sole manufacturer and will encompass the exclusive rights and responsibility of manufacture, quality assurance, and delivery.

Any tooling investment, capital or otherwise, necessary for the manufacture of this unit will also be the responsibility of Singer. This should be taken into consideration when estimating and determining the required equipment.

The estimated cost of this unit must be established within three weeks (March 26, 1964).

Under no circumstances is anyone to contact the Perma Co. other than Mr. R. A. Kloby. All written and verbal communications will be cleared through the Factory Sales Dept.

This is a confidential project and should be treated accordingly.

This control consists of four main units:

Anti-Skid device
Perma-Vac
Pressure switch
Gear box

A bill of material and a complete set of sepias (with six copies) for each of these units, plus the adapter unit, have been given to the Engineering Department.

Upon issuance of the prints, the Engineering Department will advise Mr. Romel of Sales Engineering, of the recipients so that print control can be maintained.

Mr. Sprague will be responsible for the Engineering Department's function and will assign a project leader, (Mr. J. Stein).

The prints submitted will cover everything for the unit with the exception of hoses, clamps and other hardware items.

A complete parts list is not available from the Perma Co. but Mr. Kloby will contact Mr. K. Heffer at St. Johns, Canada, in an effort to obtain one. The unit was dismantled and the parts listed at St. Johns so the reliability of the list should be sufficient without duplicating the procedure.

The Engineering Department will forward a purchased parts list with prints, to Mr. J. Moore.

Prints submitted for quotations by the Purchasing Dept. must have all references to the customer (Perma) removed.

A vendor list has been supplied and it should be noted by Purchasing that these are vendors who are not to be contacted.

Test specifications have not been determined at this time.

All work on this project is to proceed without waiting for additional information. Any information not available will be obtained and supplied at a later date.

In some instances, print tolerences and specifications have not been adhered to by the present supplier. This will be evaluated by the Engineering Department for future use.

In addition to the aforementioned, the Engineering Dept. has received:

1. A complete sensing device
2. A complete Anti-Skid control which includes:-

   (a) One static kit
   (b) One adapter kit
   (c) One variable kit

Quantities (estimated)

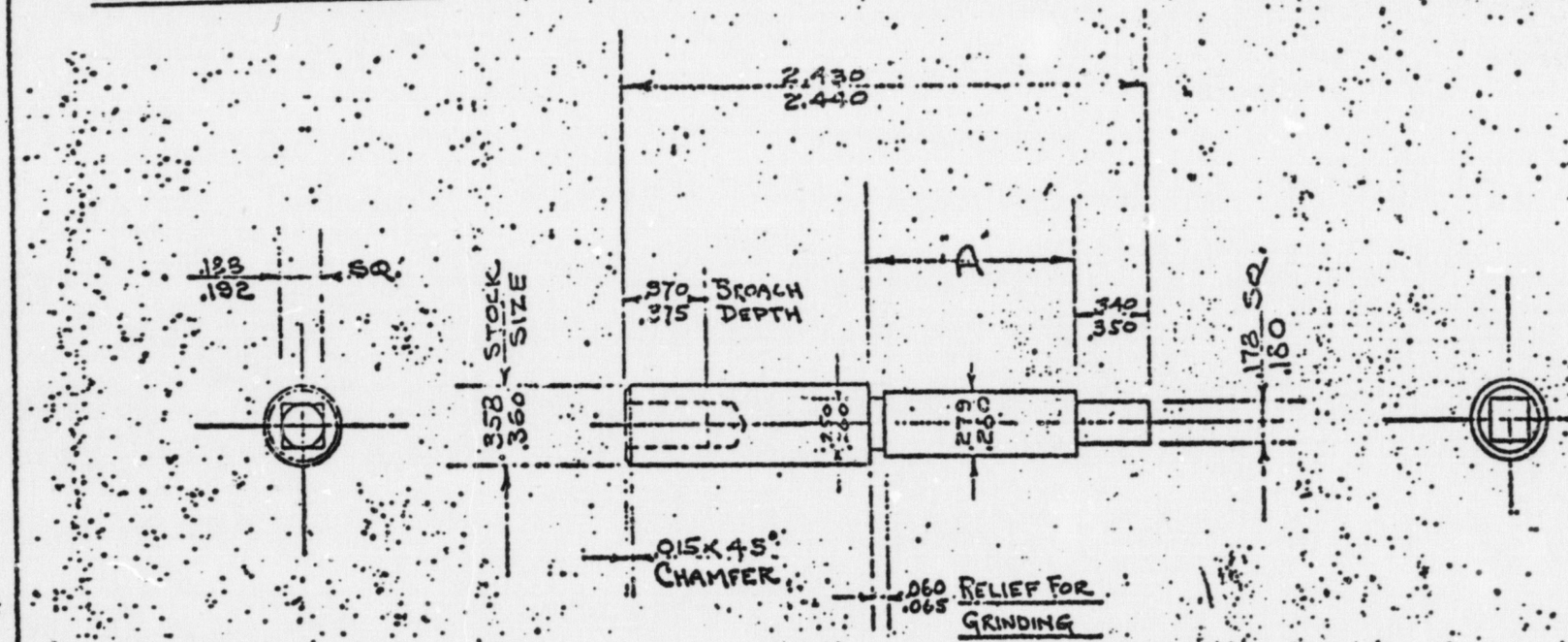
250,000/yr.
50,000/order

The next meeting (progress report) on this subject is scheduled for 10:00 a.m., March 20, 1964, in the Conference Room.

C. R. FRICKE

March 13, 1964
CRF/dh

PLAINTIFF'S EXHIBIT 23



	A	
MAKE FOR OLD BLANK ADAPTER	.960 .955	
MAKE FOR NEW BLANK ADAPTER	1.328 1.403	

CONFIDENTIAL

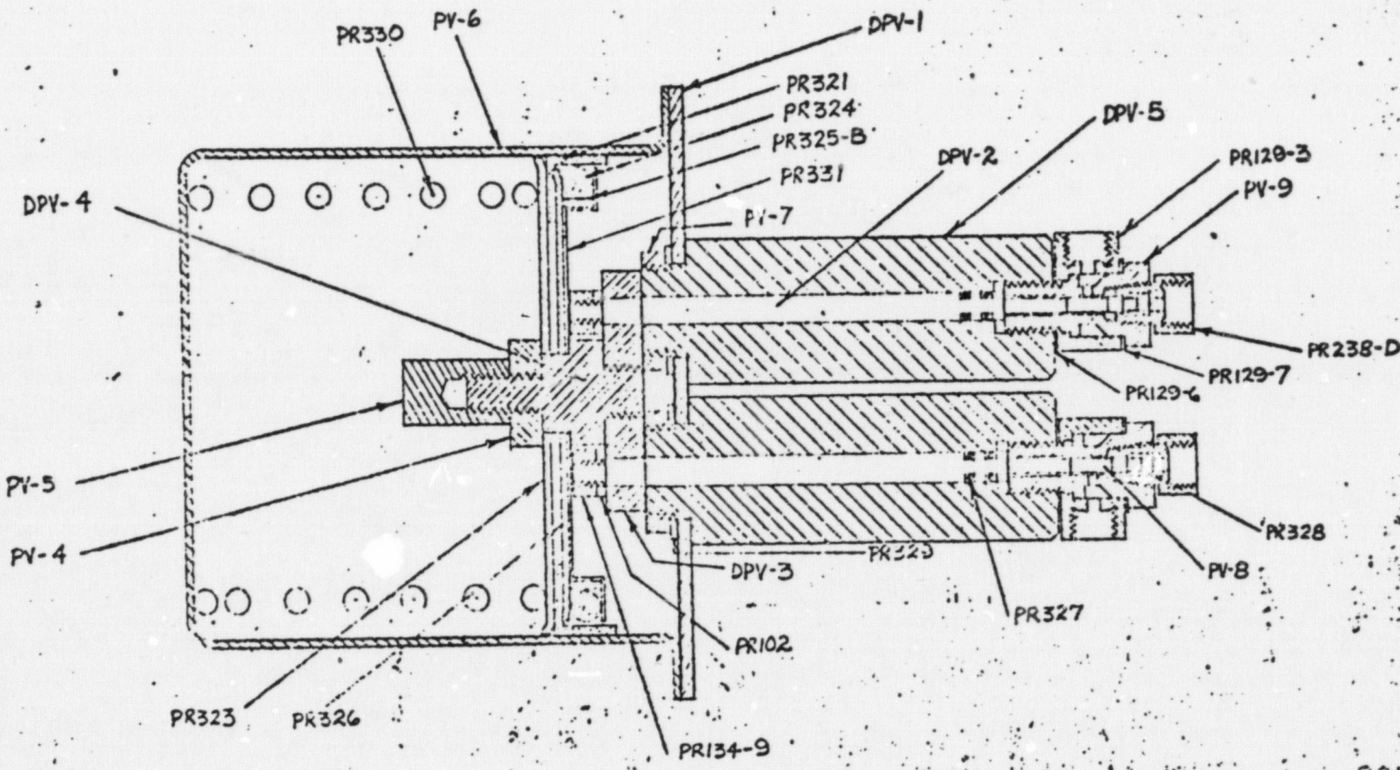
							SCALE 2X1	TITLE SPECIAL ADAPTER CONTROL
							MATL 5113	PERMA RESEARCH & DEVELOPMENT CO. NO. ATTLEBORO, MASS.
						FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	STEEL	DRAWN DAW 11/8/73
						BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	TR. BY
						2 PLACE DECIMALS ± .02 3 PLACE DECIMALS ± .010 ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH	CHK'D BY
DATE	BY	CHANGE RECORD	QT	CK		DO NOT SCALE DIMENSIONS	MISC	APPROVED

DRAWING NO. AC-1

[PX 23, p 1]

E-168 [PX 23, p 1]

[PX 23, P 2]



CONFIDENTIAL

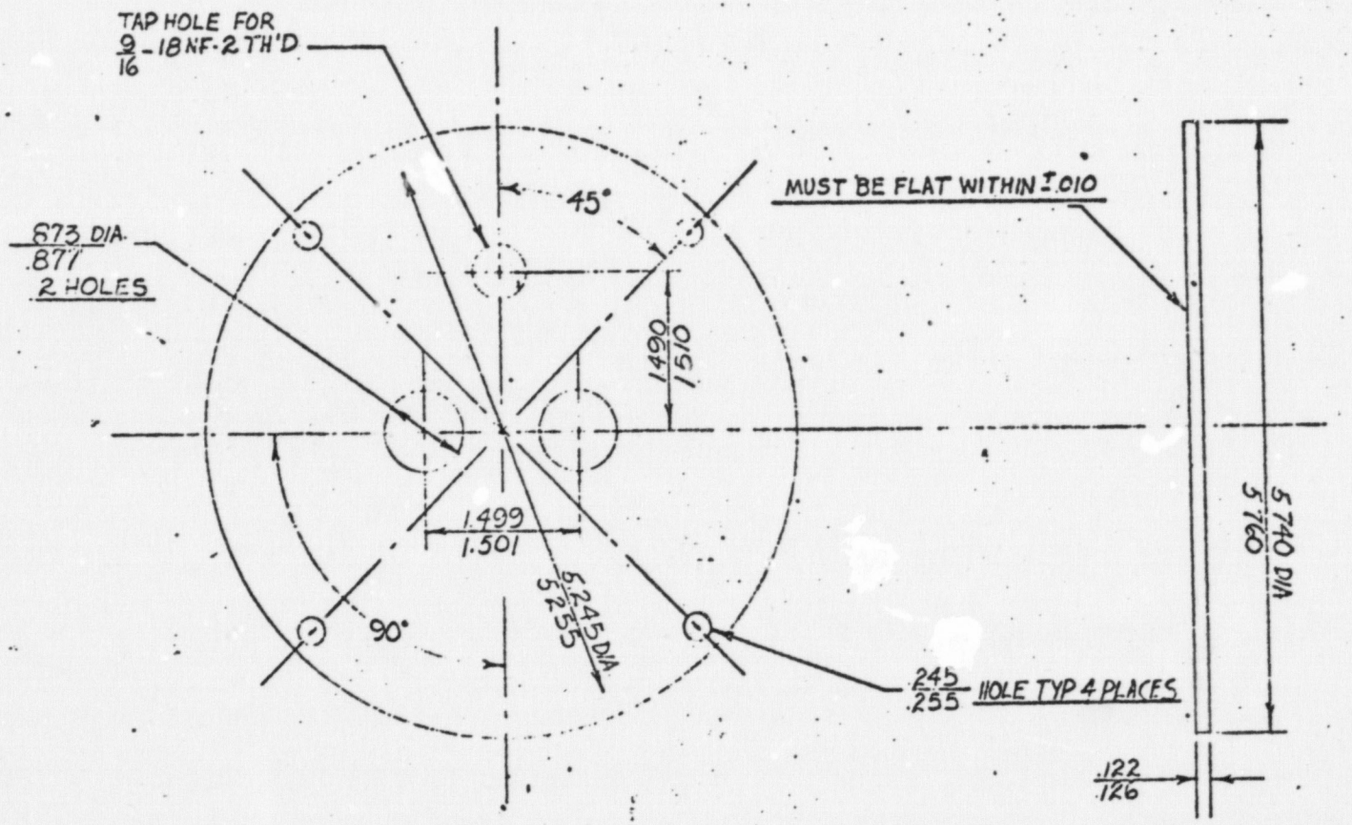
	DUAL PERMA-VAC ASSEMBLY
SCALE 1X1	PERMA-NU [illegible] CO. [illegible]
PART	[illegible] RDM 11/64
DATE	12 57
FINISH	CH D 57
MAT'L	[illegible]

DPVA

[PX 23, P 2]

E-169

[PX 23, p 3]

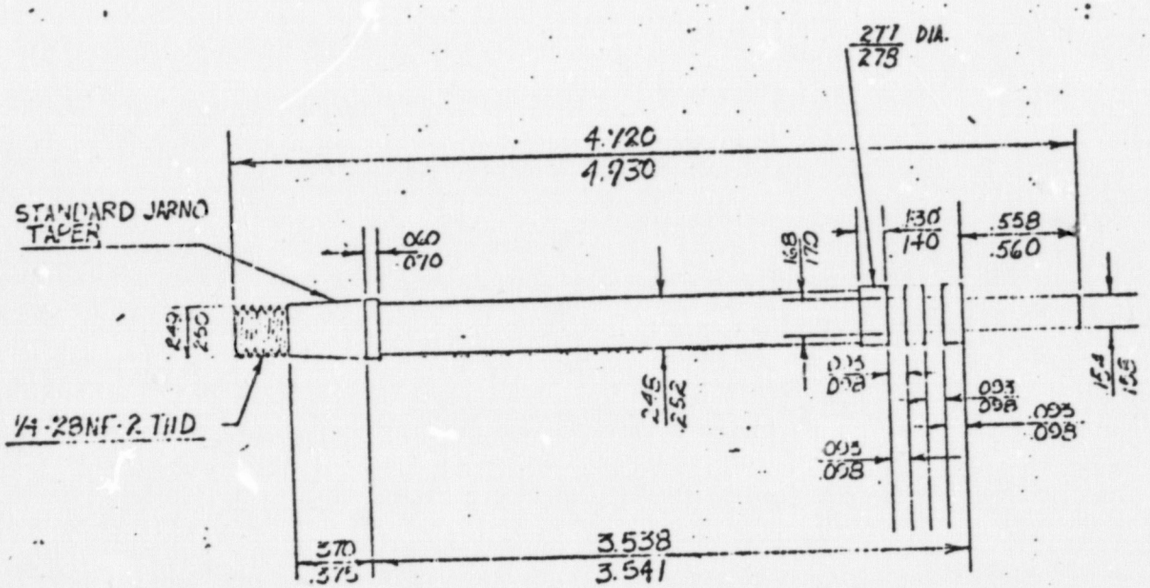


-NOTE: MATERIAL - AISI 1010 H.R.S.
ZINC - CHROMATE

	TITLE COVER FOR DUAL PERMA-VAC	
SCALE ACTUAL	PERMA RESEARCH & DEVELOPMENT CO	
MATL SEE NOTE	DR BY R D M	DATE 2/10/64
DATE	TR BY	
FINISH	CHK BY	
REQ'D 1	APPTS	DPV-1

[PX 23, p 3]

[PX 23, P 4]



NOTE: MATERIAL - COLD ROLLED STEEL MACH-S
BREAK ALL CORNERS .015 MAX.

CONFIDENTIAL

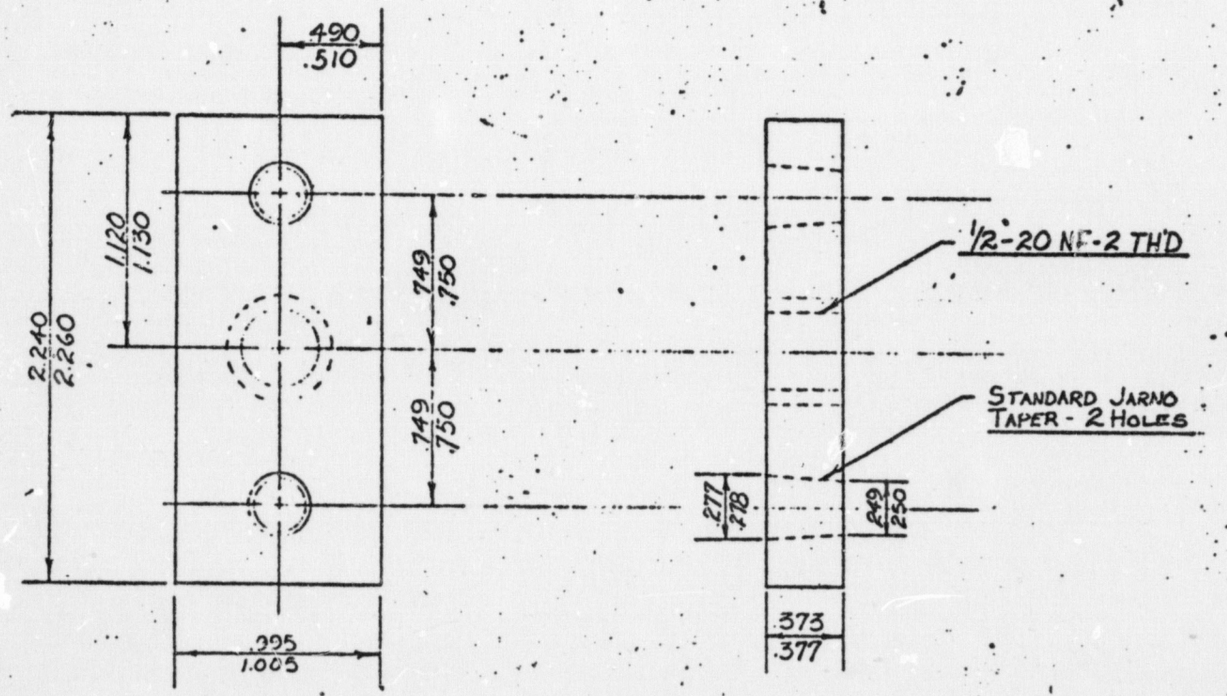
	TITLE PISTON FOR DUAL PERMA W
SCALE 2X1	PERMA RESEARCH LABORATORIES
MAT'L SEE NOTE	DRAWN BY BDM 2/19/54
FINISH	CHECKED BY
ASSY 2	APPROVED

DPV-2

E-171

[PX 23, P 4]

[PX 23, p 5]



NOTE: MATERIAL — CRS.

CONFIDENTIAL

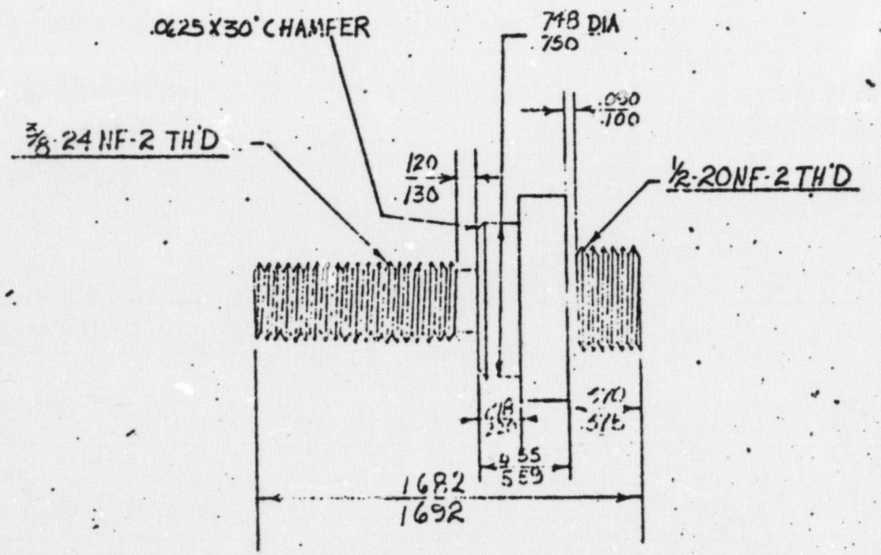
	TITLE YOKE FOR DUAL PERMA-VAC
SCALE 2X1	PERMA POWER & DEVELOPMENT CO. US RTE 1 NORTH ATTLEBORO MASS.
MAT'L SEE NOTE	DWN BY P.D.M. 3/1/65
FINISH	CHD BY
FILED 1	APPD BY

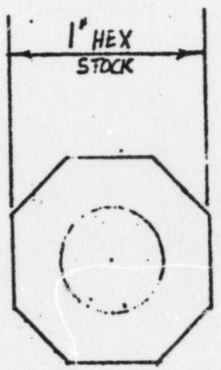
DPV-3

2-172

[PX 23, p 5]

[PX 23, p 6]





NOTE:
MATERIAL — C.R.S. B1113
BREAK ALL CORNERS .015 MAX.

CONFIDENTIAL

	TITLE ASSEMBLY SCREW FOR DUAL P.V.
SCALE 2X1	PERMA RESEARCH & DEVELOPMENT CO.
MATL SEE NOTE	DR'N BY RDM
FINISH	TR'D BY
	CH'D BY
REQ'D 1	APPROVED

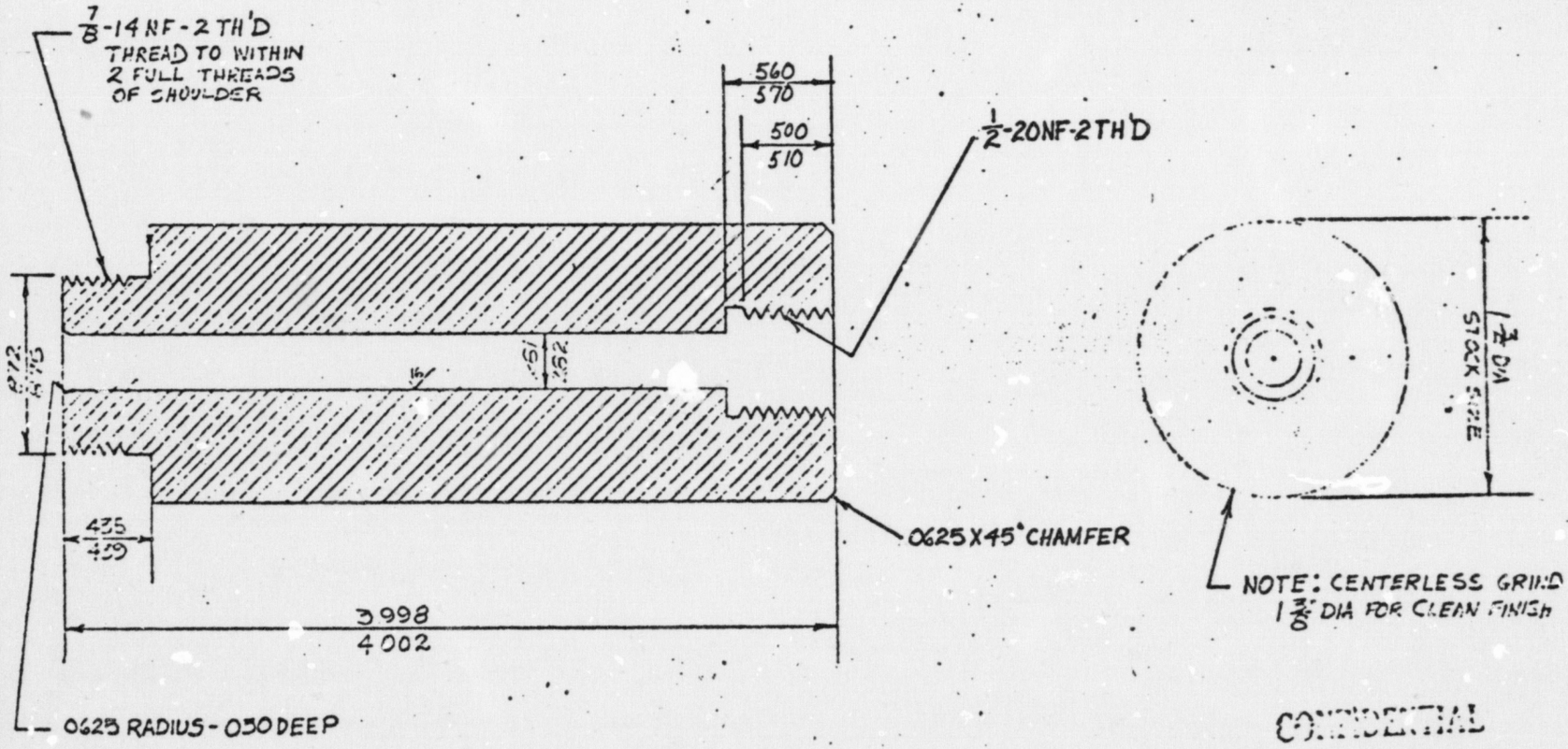
DPV-4

E-173

[PX 23, p 6]

[PX 23, p 7]

NOTE: MATERIAL 2011-T3 ALUMINUM



CONFIDENTIAL

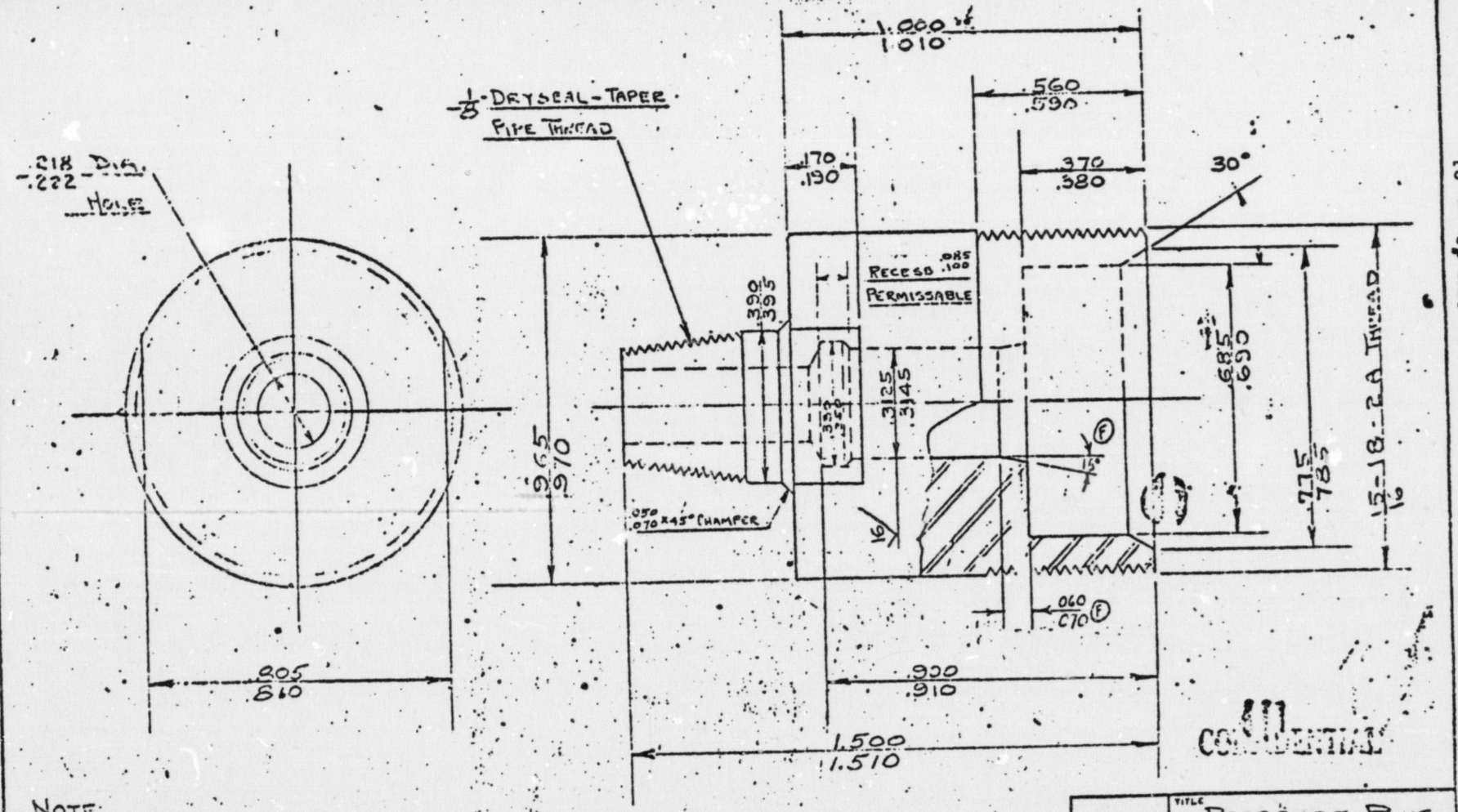
	PISTON BODY FOR DUAL MECHANICAL
SCALE 2X1	FERNA RESEARCH-DEVELOPMENT 345 E WASHINGTON NATTLEBORO MASS
MAT'L ALUMINUM 2011-T3	DRAWN F.D.H. 1/12/54
FINISH	CHECKED
REQ'D 2	APPROVED

DPV-5

[PX 23, p 7]

E-174

[PX 23, p 8]



NOTE:
CADMIUM PLATE .0003 MIN.
BEFORE BURNISHING .313 / .315 DIA HOLE

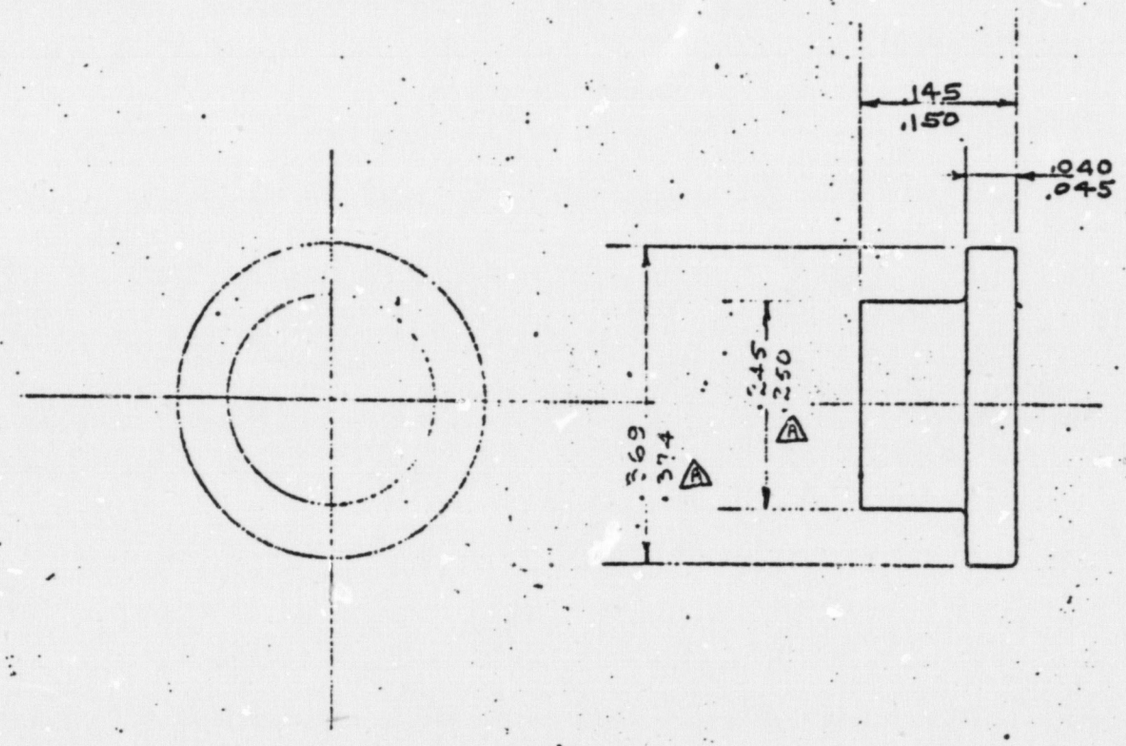
10-21-64	F	45° TAIL CHAMFER TOLERANCE				SCALE 4 X 1	TITLE PRESSURE PLUG
4-11-62	E	CHANGE TAIL TO 1/8" FLAT			FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	MAT'L B 1113	
2-4-62	D	PLATING			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	DRAWN L. DATE
11-20-61	C	REVISION			ALLOWANCE ON FUNCTIONAL DIMENSIONS	FINISH	CHK'D
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D	APP'D

DRAWING NO. PS-6F

[PX 23, p 8]

E-175

[PX 23, p 9]

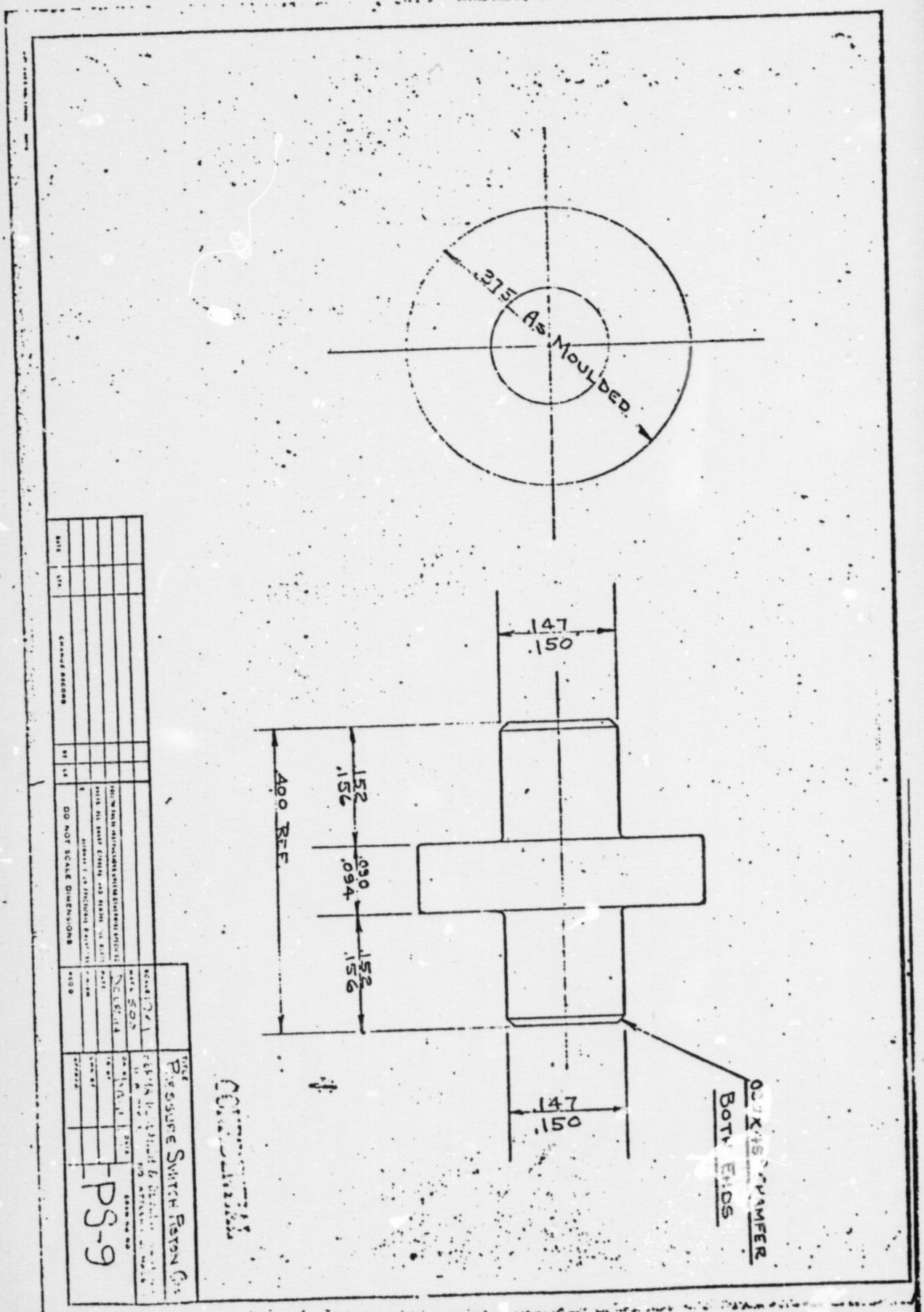


NOTE:
CADMIUM PLATE
.0002 MIN. THICKNESS

						SCALE 2X1	TITLE SPRING RETAINER
						DATE 6-5-63	FERRIS RESEARCH & DEVELOPMENT CO.
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	APPD.	DRAWING NO. PS-7
					BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	DATE	
11-25-63	A	DIM WERE INTERCHANGED	DK		X DECIMALS ON FRACTIONAL DIMENSIONS	FINISH	
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MATL	

[PX 23, p 9]

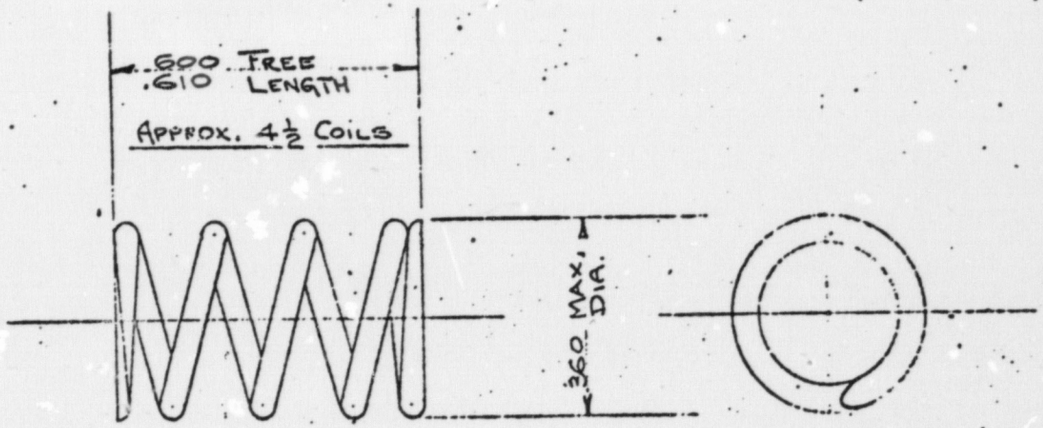
E-176


.215
As Moulded
.147
.150
.152
.156
.090
.094
.152
.156
.490 REF
.147
.150
DO NOT SCALE DIMENSIONS
PS-9

[PX 23, P 11]

[PX 23, P 11]

E-178



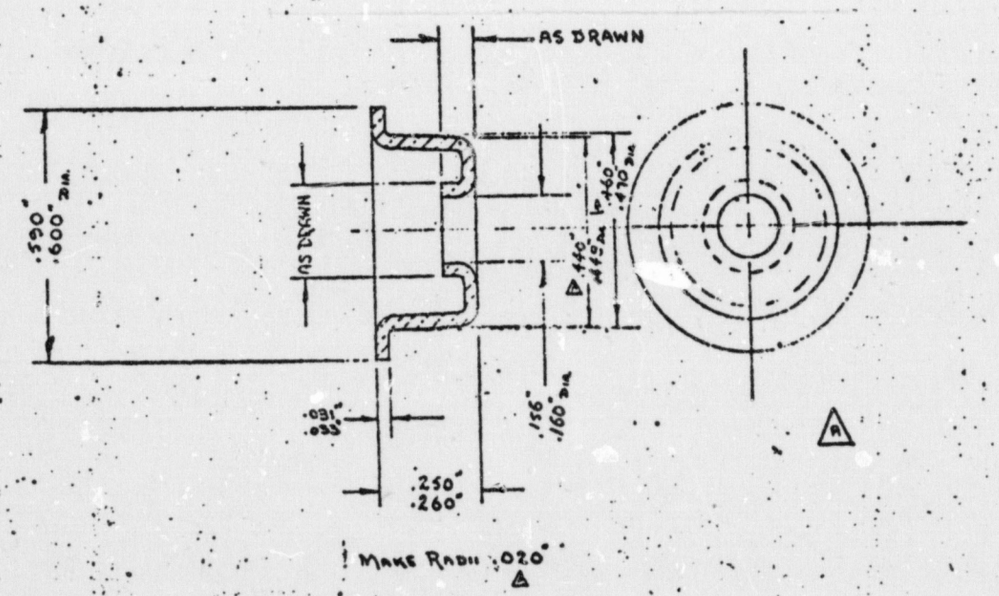
NOTE:

MAKE FROM .045 DIA MUSIC WIRE - HARDEN & TEMPER. WITH CLOSED & GROUND ENDS. - SPRING TO HAVE FORCE OF $14.8^{+}_{-}{}^{.5}_{.5}$ OZ WHEN COMPRESSED TO $\frac{.375}{.350}$

CADMIUM PLATE & REMOVE HYDROGEN EMBRITTLEMENT

						SCALE 5X1	TITLE PRESSURE SWITCH SPRING
						MATERIAL	
						PART	
						FINISH	
		TOLERANCES			DO NOT SCALE DIMENSIONS	WEIGHT	DRAWING NO. PS-5
DATE	LTR	CHANGE RECORD	BY	CK			

[PX 23, p 12]



NOTE:
SILVER PLATE .0005" MIN.

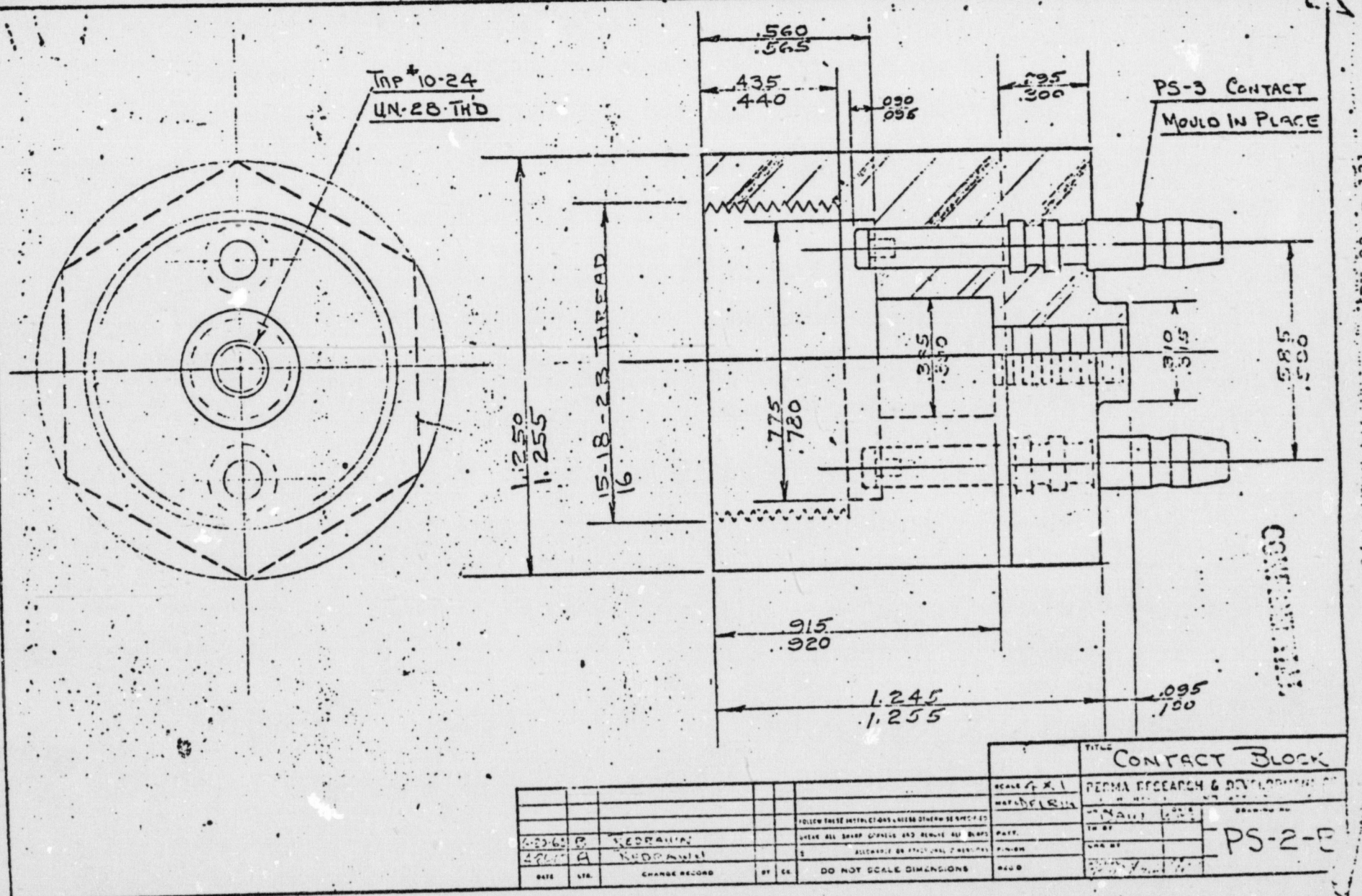
CONFIDENTIAL

					CONTACT RING
					4 X 1
					BRASS
11-25-63	A	Re-Drawn & Dim. Changed			
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DRAWINGS

[PX 23, p 12]

E-179

[PX 23, p 13]



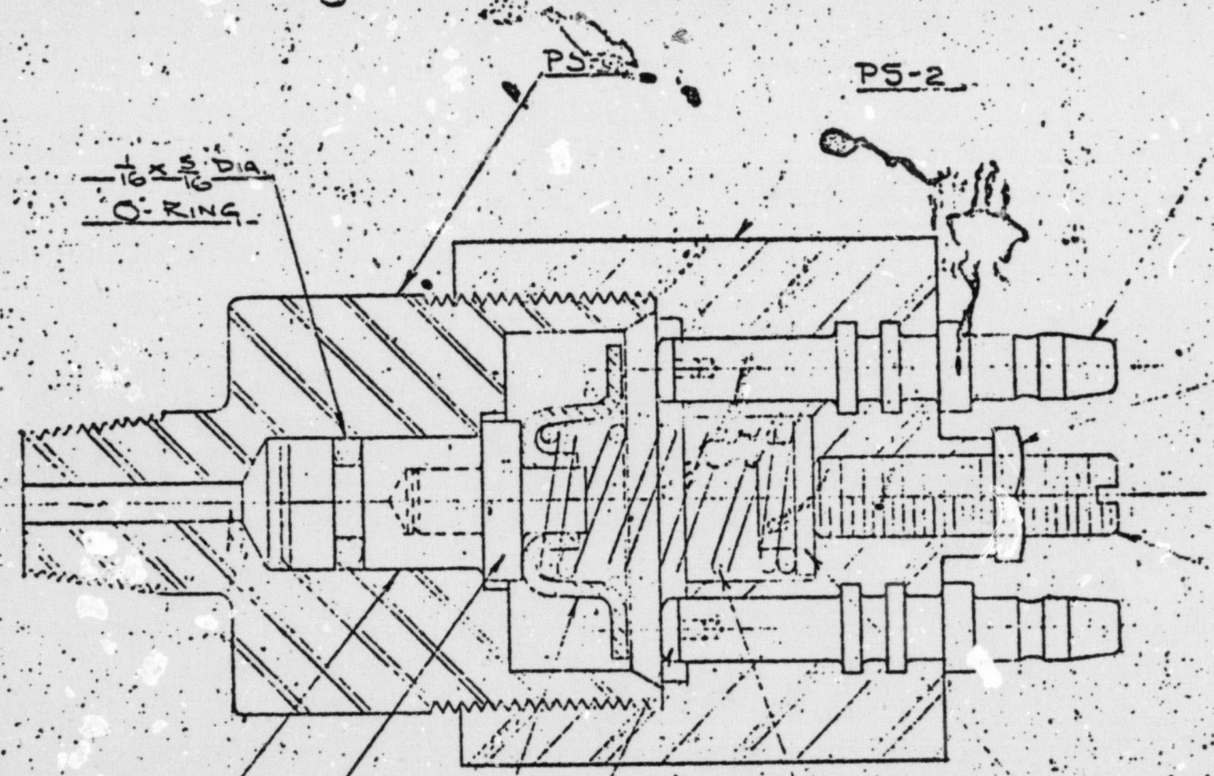
[PX 23, p 13]

E-180

[PX 23, p 14]

[PX 23, p 14]

E-181



CONFIDENTIAL

PATENT PENDING

DATE	LTR.	CHANGE RECORD	BY	CK
6-26-63	A	ADDED ASSEMBLY TOOL		
		DO NOT SCALE DIMENSIONS		

SCALE 2 X 1

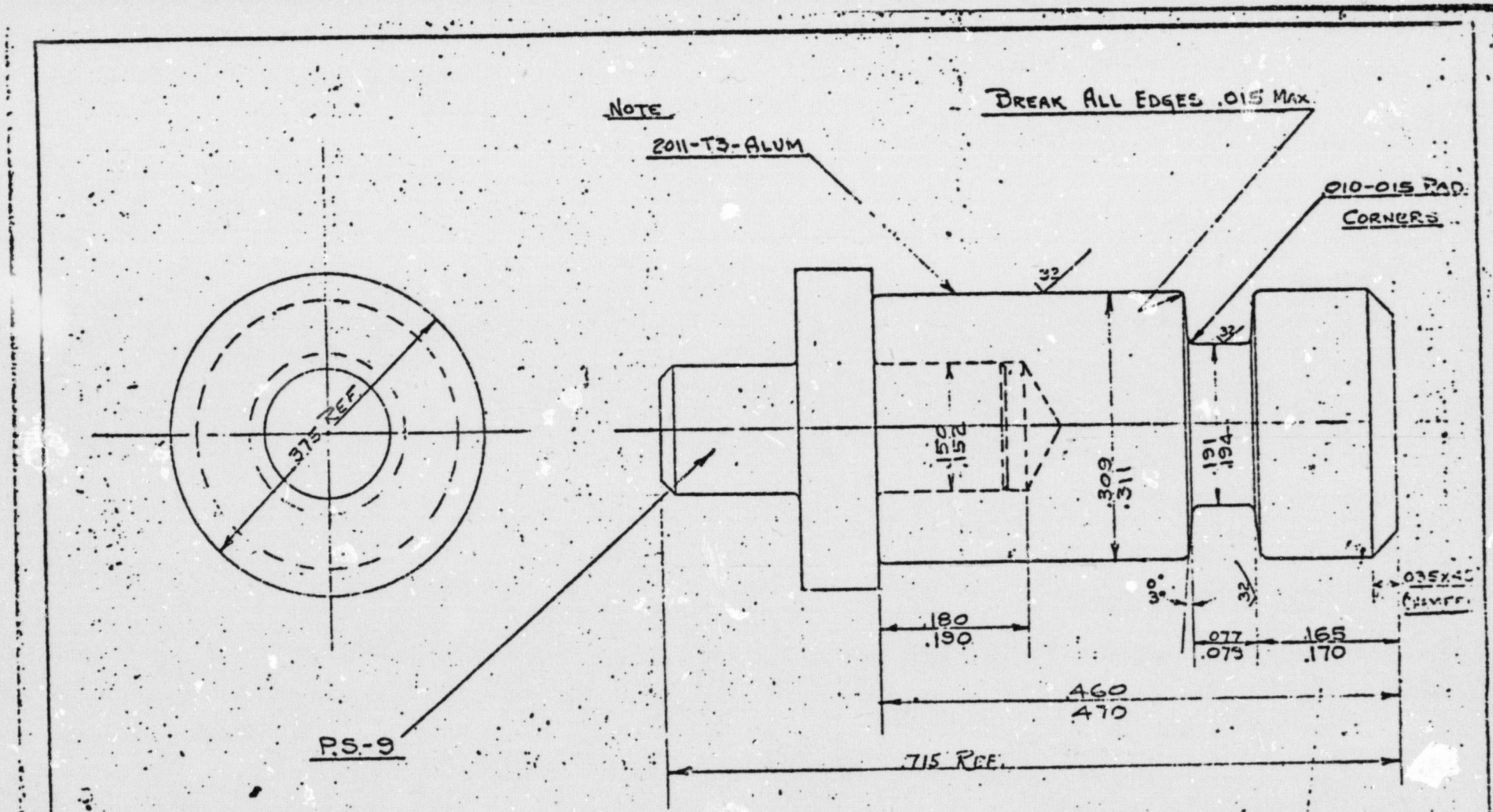
MAT'L

FINISH

TITLE ASSEMBLY PRESSURE SWITCH

DRAWING NO. PS-1

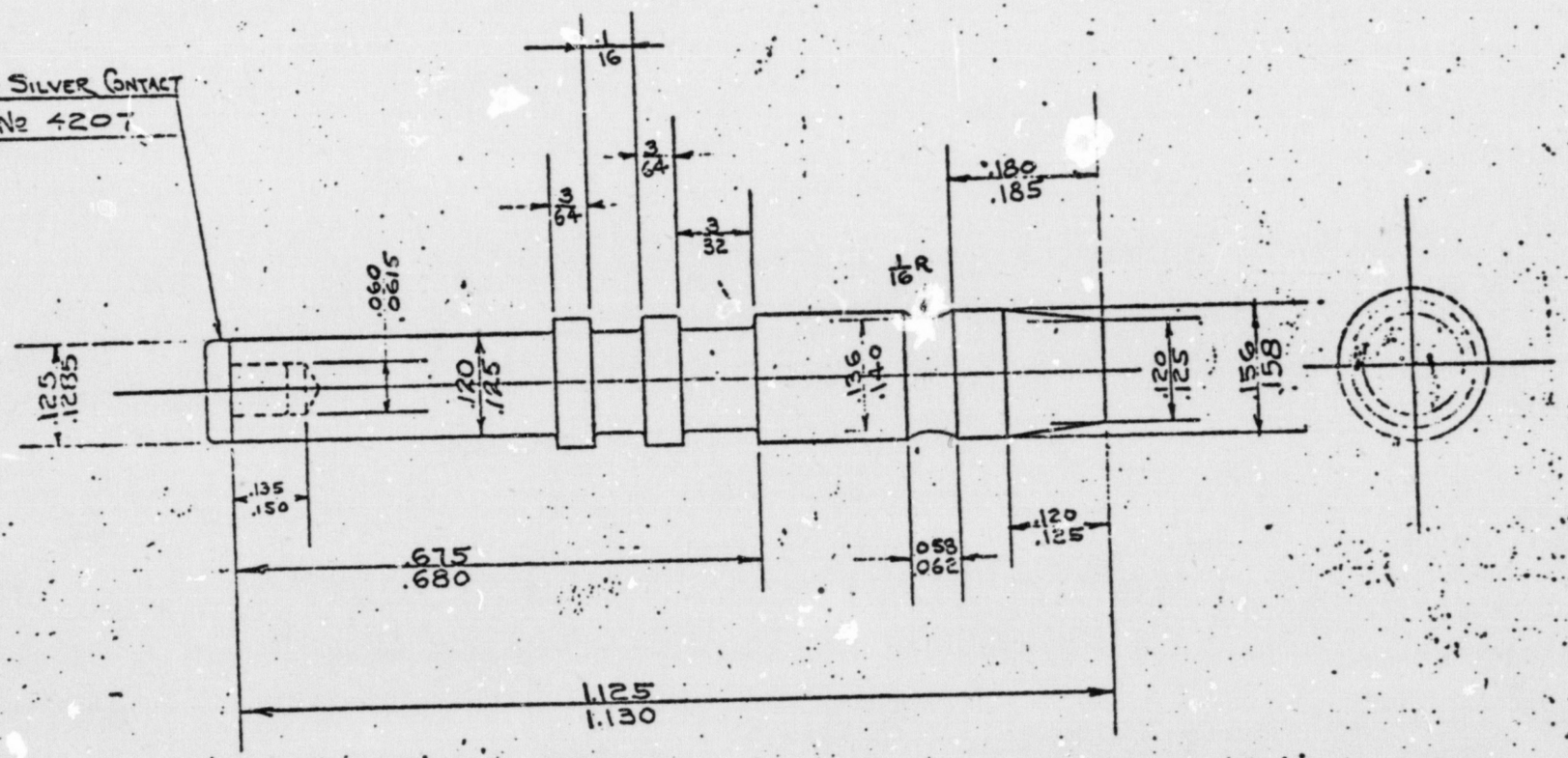
[PX 23, p 15]



[PX 23, p 15]

E-182

M.S.C. - SILVER CONTACT
PART No 4207



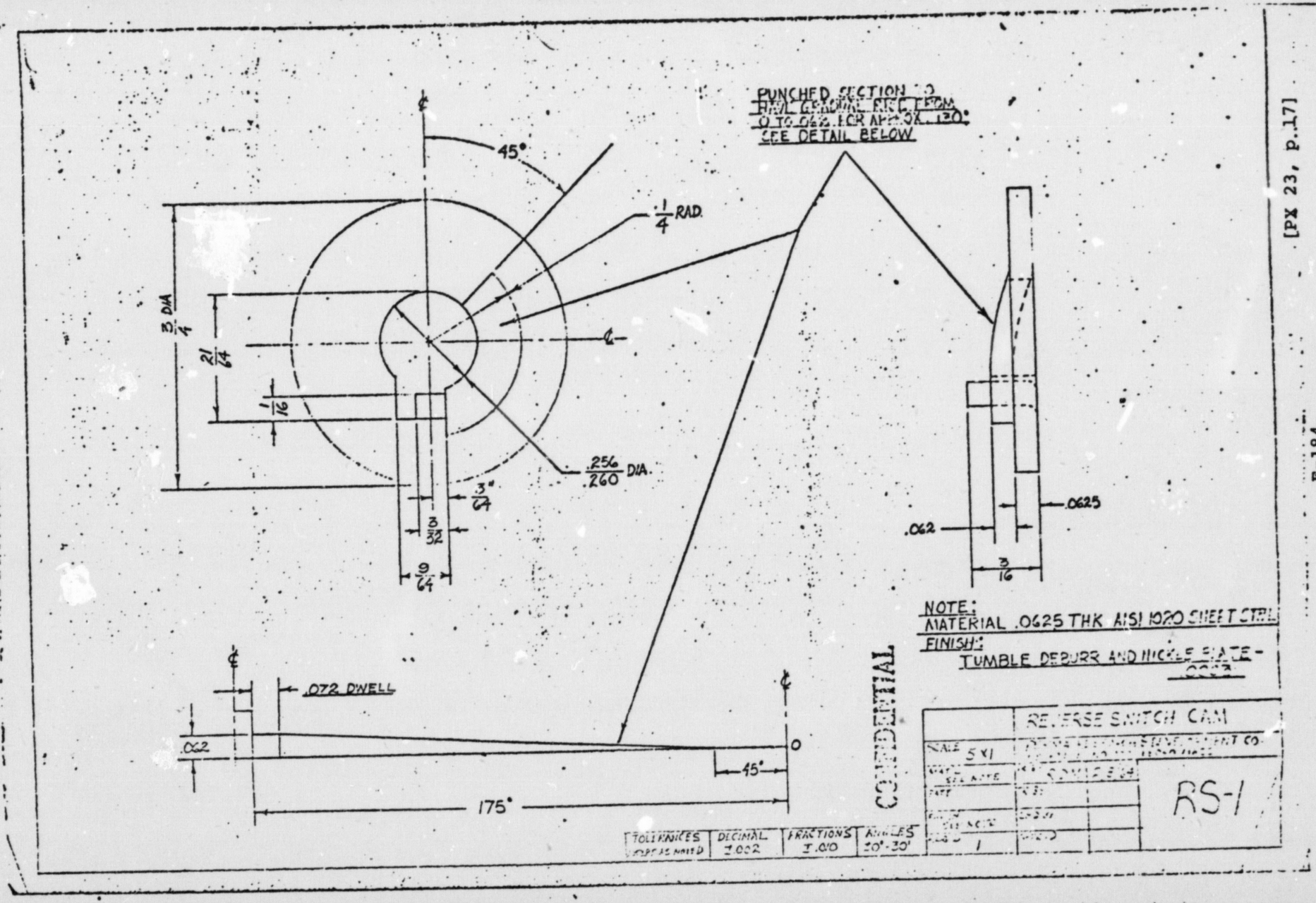
CONFIDENTIAL

NOTE:
SILVER PLATE .0003 MIN.

				SCALE 8 X 1	TITLE PRESSURE SWITCH CONTACT
				DATE 2-1-65	CEMA RESEARCH & DEVELOPMENT CO
				FOLLOW SHOP INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	DR BY [illegible] DATE
				BREAK ALL SHARP EDGES AND REMOVE ALL BURRS	PATT. TR BY
2-16-65	D	REDRAWN		± .005 ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH LOG BY
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D 2	APPROVED [illegible] DATE

DRAWING NO. P5-3D

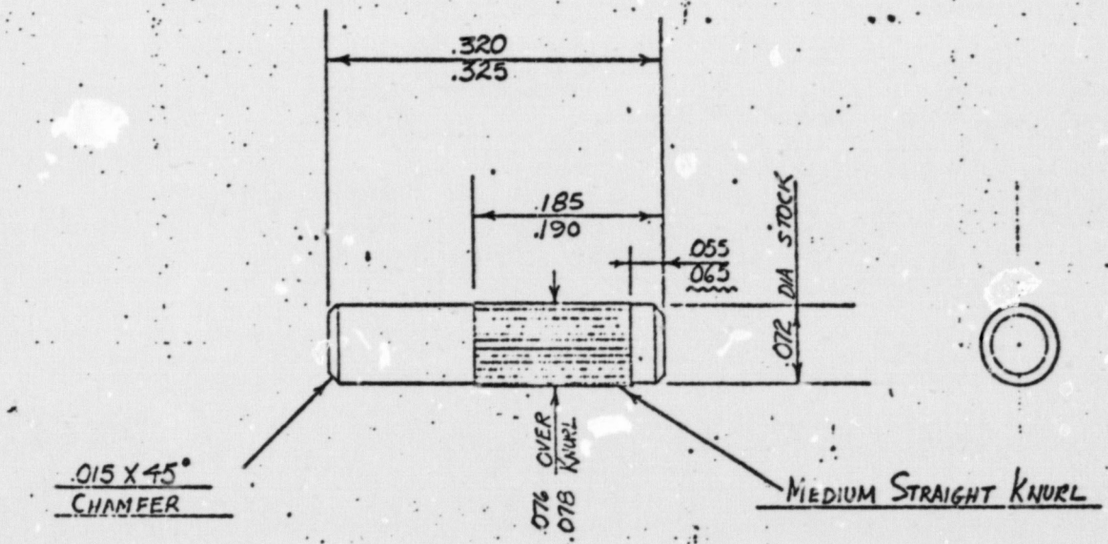
[PX 23, p 17]



[PX 23, p.17]

E-184

[PX 23, p 18]



NOTE

HARDEN & DRAW TO
C-45-47 ROCKWELL
NICKEL PLATE .0003 MIN.

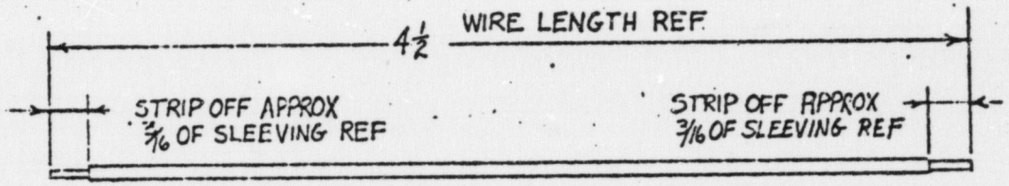
CONFIDENTIAL

	TITLE CAM WASHER SHAFT PIN		
SCALE 10X1	PERMA RESEARCH AND DEVELOPMENT CO		
MATL 1020 STEEL	DRN RDM	DATE 1/10/54	RS-2
HEAT TR	CHK		
FINISH	APPD		
QTY 1			

[PX 23, p 18]

E-185

[PX 23, p 19]



NOTE:

*20 GA WHITE WIRE LEAD. SOLDER ONE END OF WIRE TO LOWER FLAT CONTACT AND OTHER END OF WIRE TO CONTACT HAT IN ASSEMBLY
WHITE WIRE LEAD IS 1/32 WALL THERMO-PLASTIC. INSULATED

CONFIDENTIAL

	WIRE LEAD FOR REVERSE SWITCH
SIZE 2X1	PERMA RESEARCH & DEVELOPMENT CO
MAT'L #20 GA WIRE	RDW
DATE	
FINISH	
SHEET 1	

RS-3

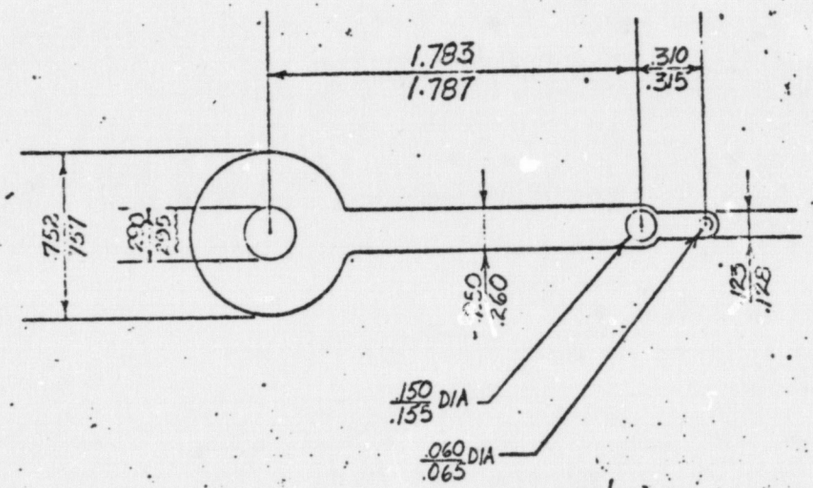
[PX 23, p 19]

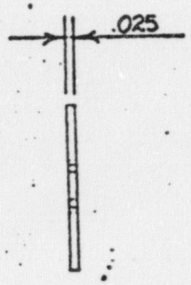
[PX 23, p 20]

[PS 23, p 20]

E-187

NOTE:
MATERIAL GILDING #4 BRASS ½ HARD





CONFIDENTIAL

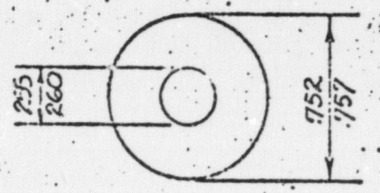
	TITLE THRUST WASHER WIRE CONTACT
SCALE 2X1	PERMA RESEARCH & DEVELOPMENT CO
MAT'L SEE NOTE	DR BY RDM
PATT	CK BY
FINISH	APVD BY
REQ'D	

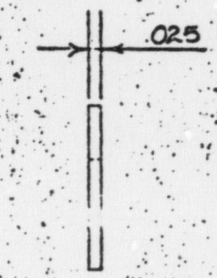
RS-4B

[PX 23, p 21]

NOTE:

MATERIAL GUILDING #9 BACS
½ HARD





CONFIDENTIAL

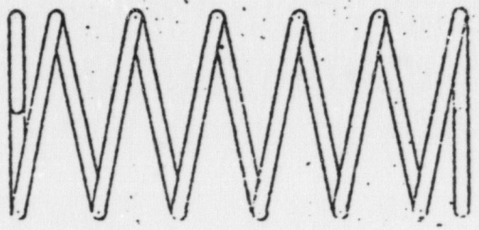
	SPRING THRUST WASHER
SCALE 2X1	
FINISH SEE NOTE	

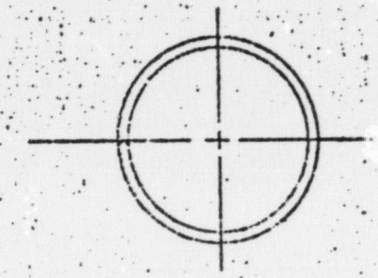
RS-5

[PX 23, p 21]

E-188

[PX 23, P 22]





NOTE:

DIAMETER OF WIRE	.018
OUTSIDE DIMENSION OF SPRING	.240-.250
ACTIVE COILS	5
TOTAL COILS	7
DIRECTION OF HELIX	OPTIONAL
FREE LENGTH	.510-.520
LOAD AT COMPRESSED LENGTH OF .250 INCH	5 OZ. ± 5%

ENDS SQUARED, NOT GROUND
MATERIAL: .018 PHOSPHOROUS BRONZE

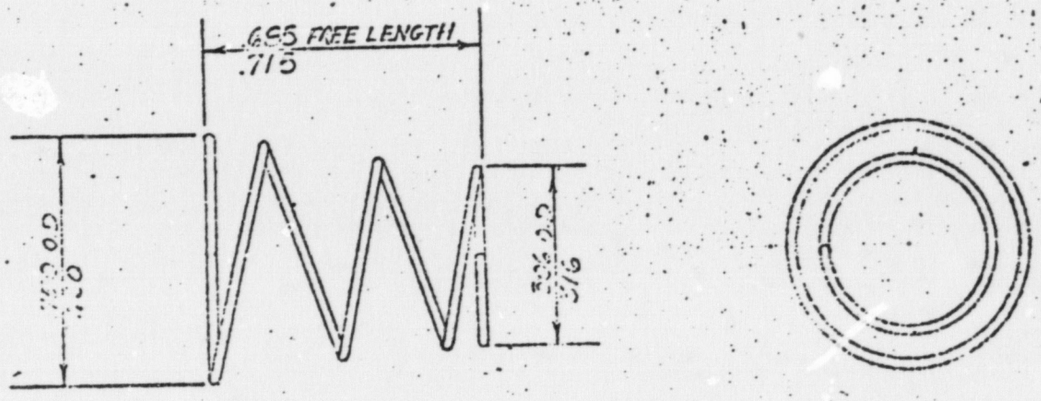
CONFIDENTIAL

SIZE 8x1	TITLE CONTACT SPRING
DATE SEE ENG.	
	RS-7

[PX 23, P 22]

E-189

[PX 23, p 23]



NOTE:

DIAMETER OF WIRE	.026
NORMAL O.D.	.306-.316
TOTAL COILS	4
DIRECTION OF HELIX	OPTIONAL
FREE LENGTH	.625-.715
LOAD AT MINIMUM COMPRESSED LENGTH	11 oz. ± 5%
TOTAL ACTIVE COILS	2

MATERIAL: PHOSPHOROUS BRONZE
ENDS SQUARED, NOT GROUND

CONFIDENTIAL

SCALE 4X1	TITLE [illegible]
DATE OF DWG.	MODEL
	PS-6

[PX 23, p 23]

E-190

[PX 23, p 24]

[PX 23, p 24]

E-19

NOTE:

MATERIAL: BRASS

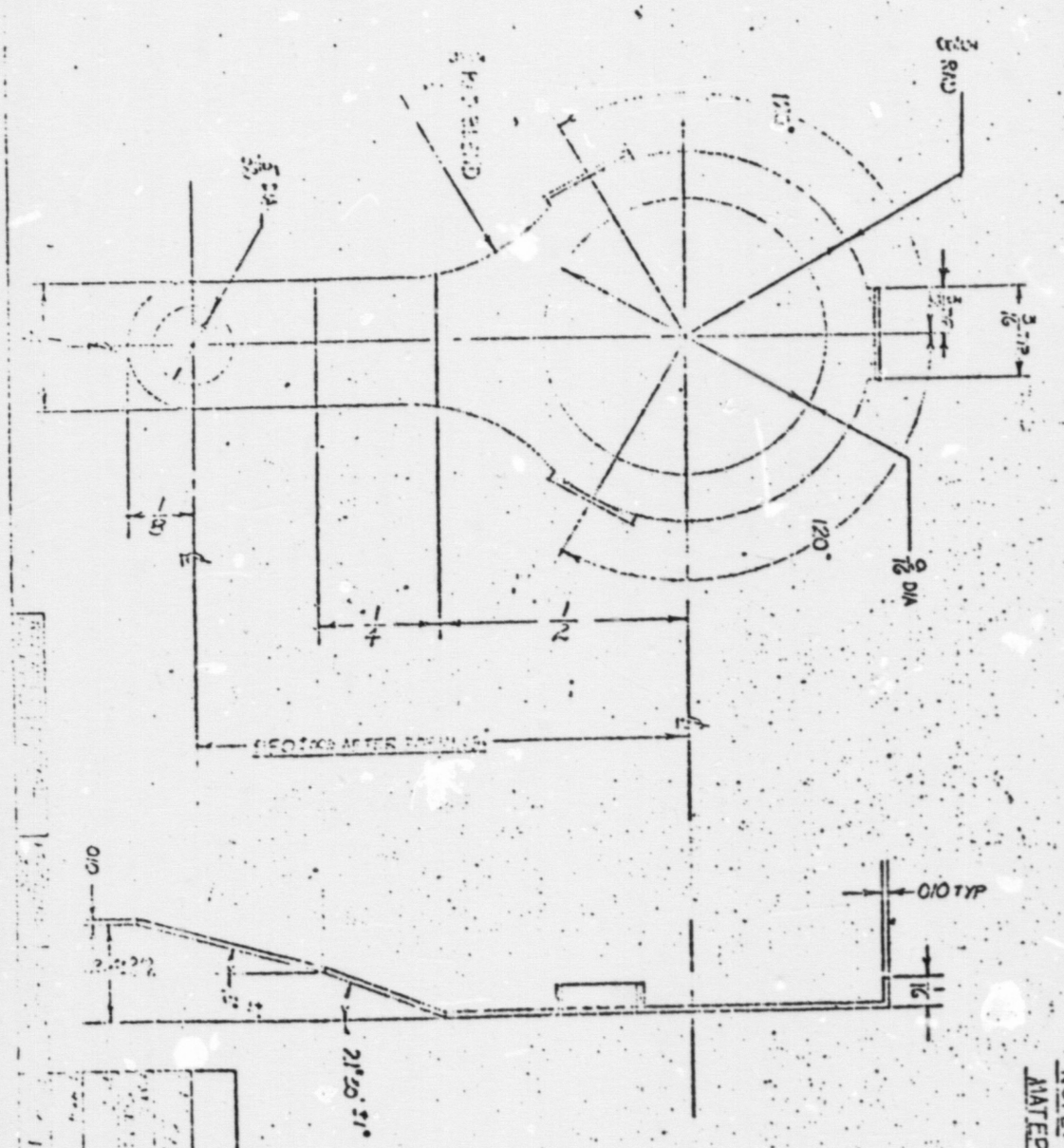
LEAVE NO BURRS OR

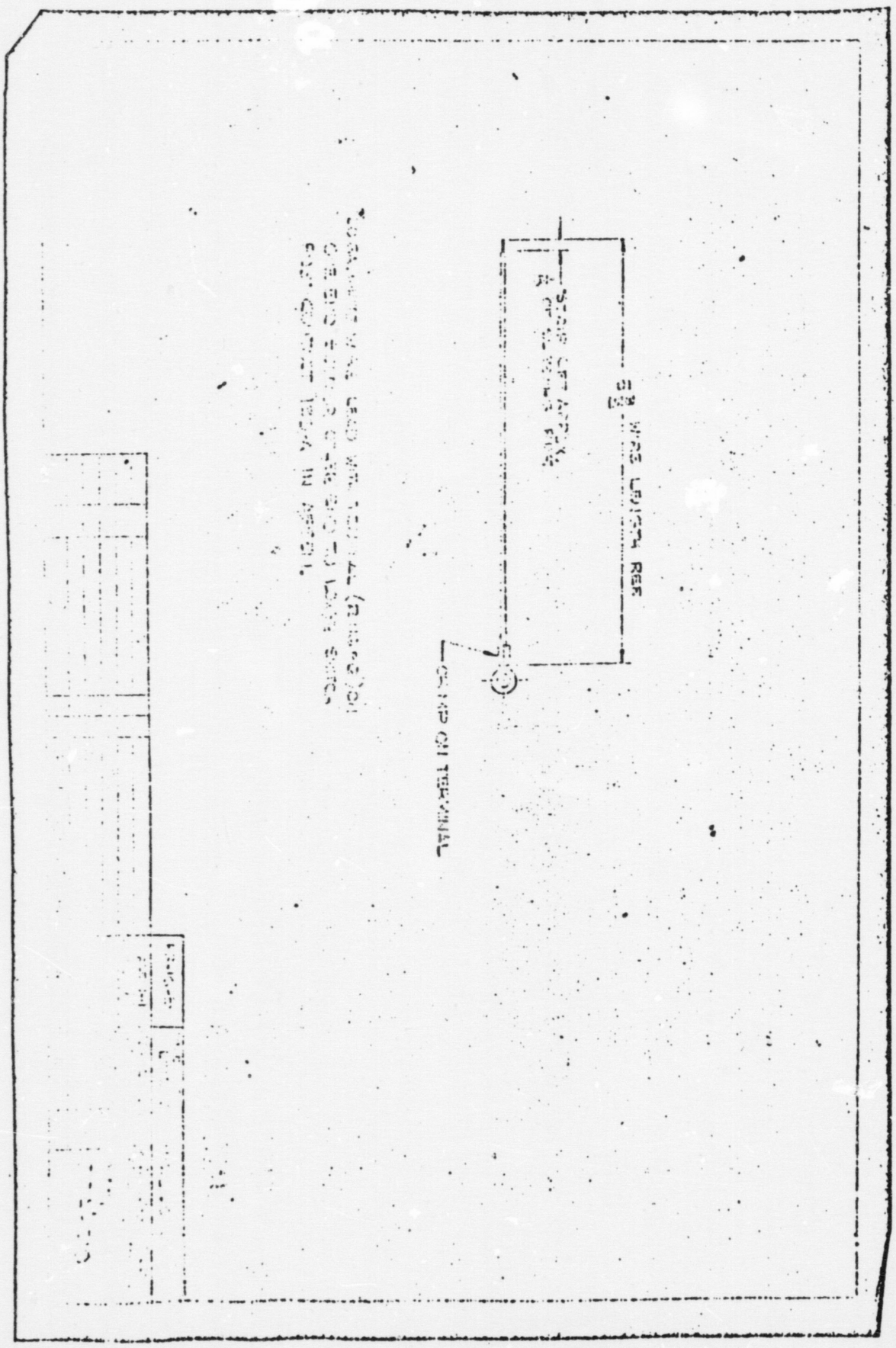
SHARP EDGES

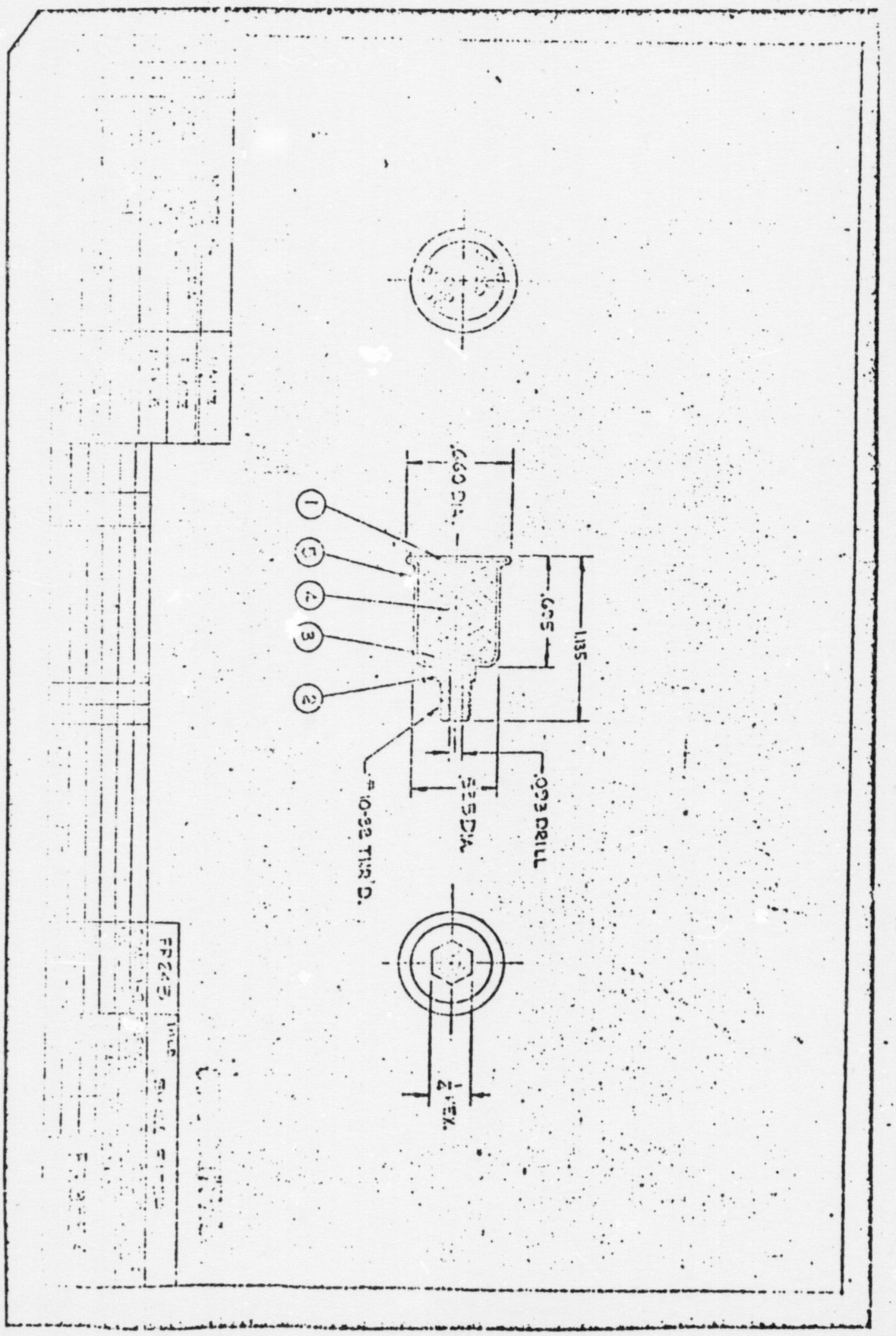
SQUARE CORNERS REQUIRED .000 TO .005 MAX BREAK

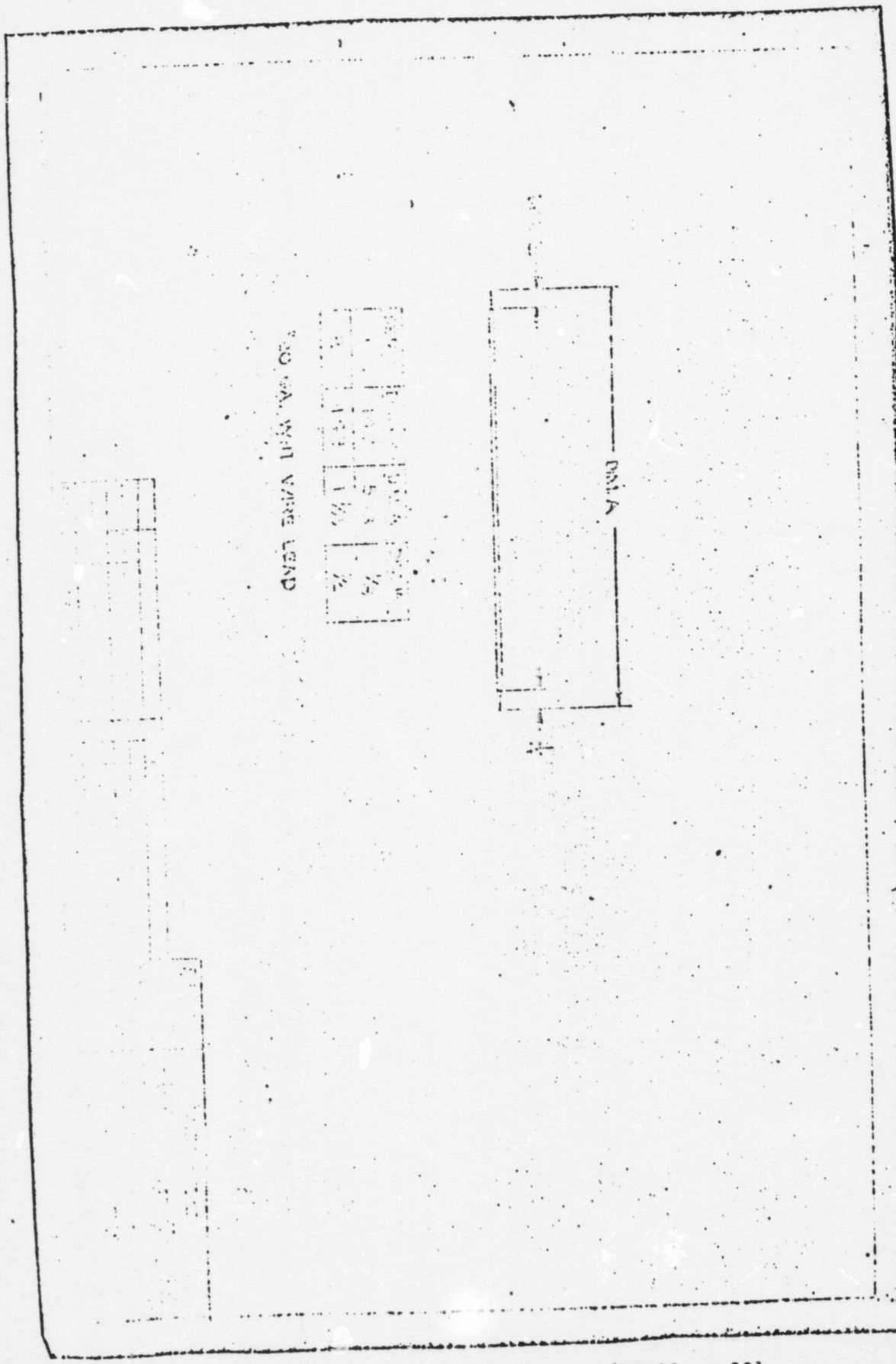
CONTRACT

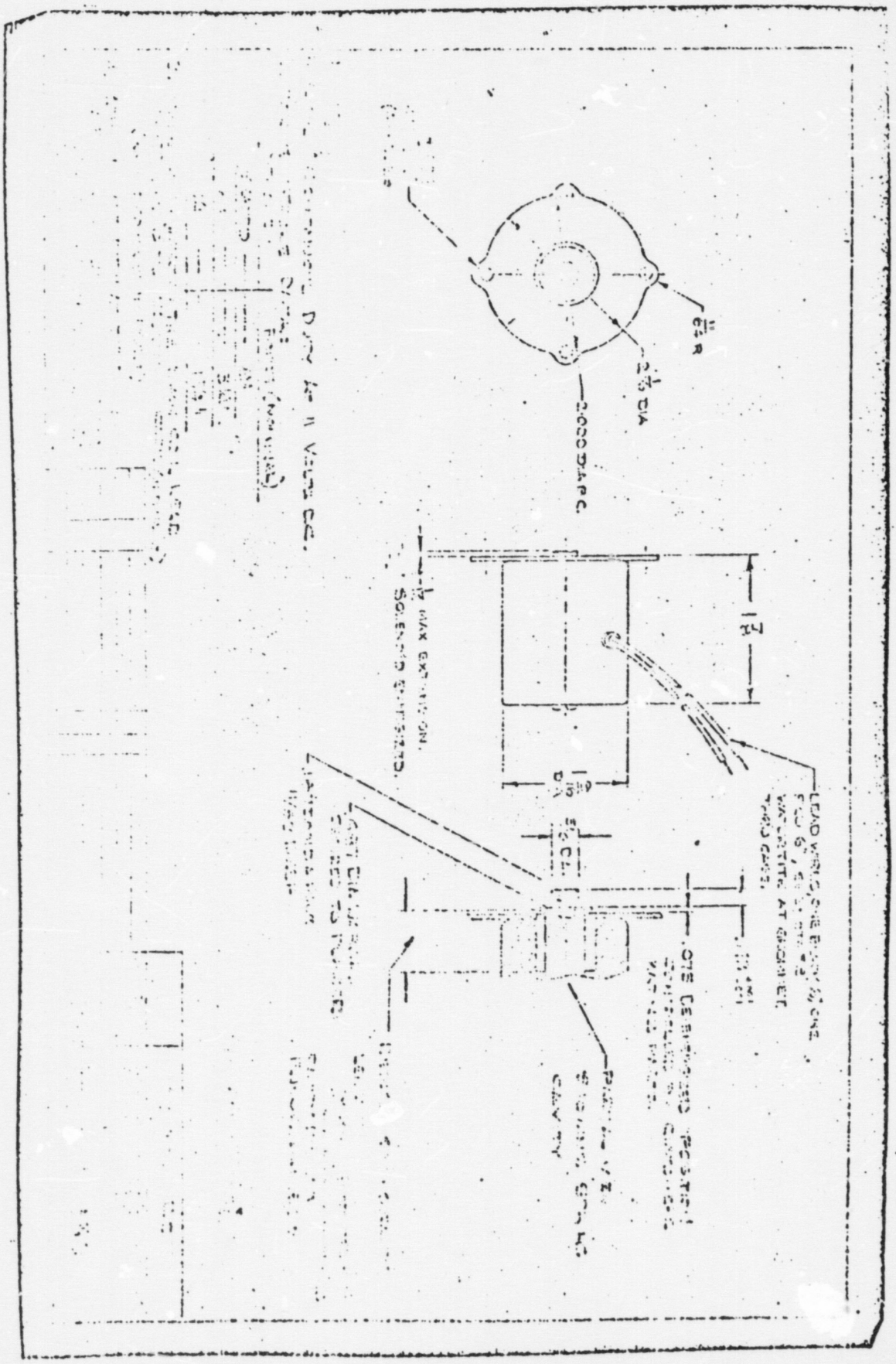
	TITLE CONTACT	
		RS-0

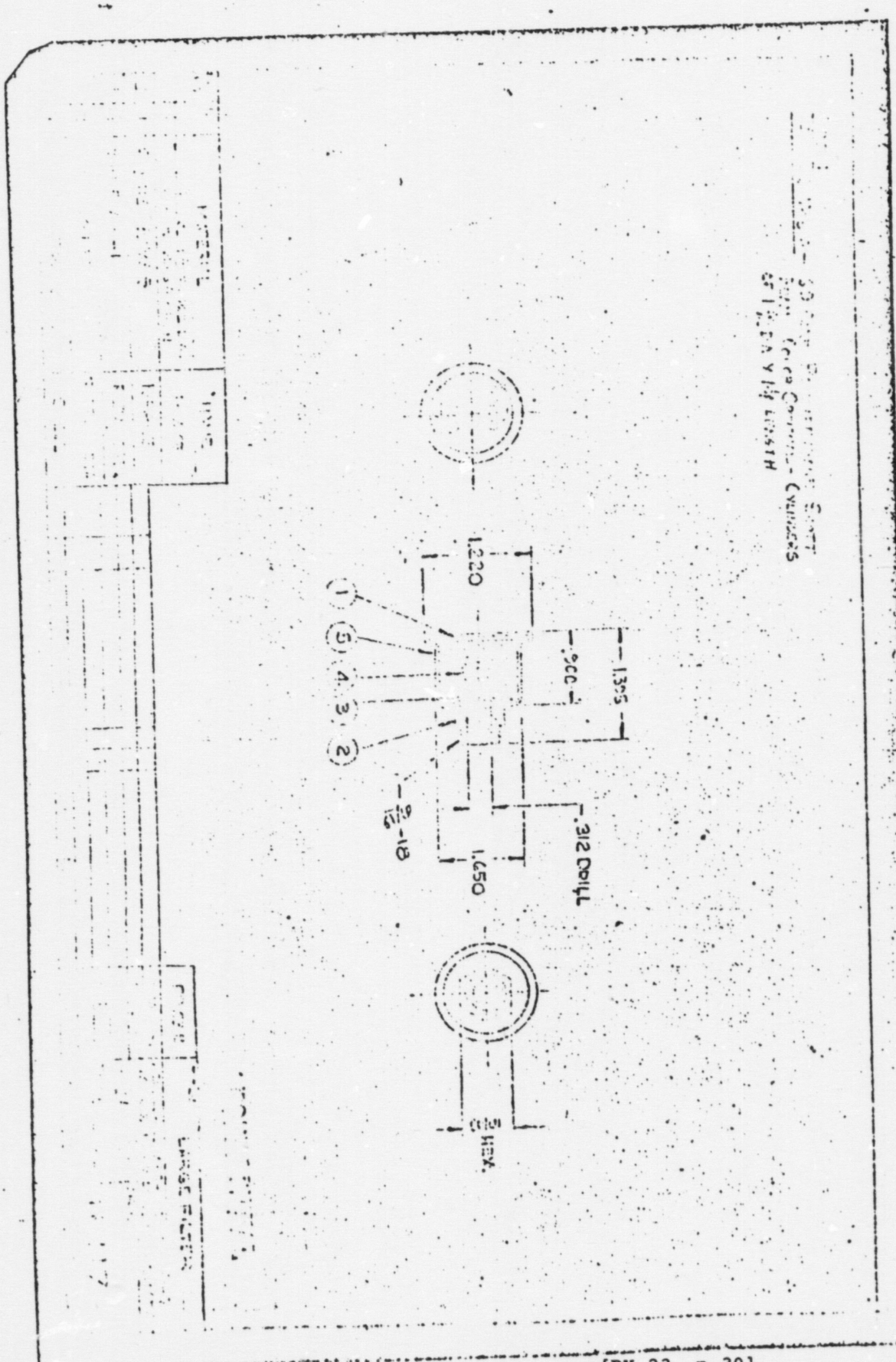

120°
120°
120°
.010 TYP
21°30'
NOTE:
MATERIAL #30(.010) PHOS. BRONZE
½ HARD




.660 DIA.
.635
1.135
#10-32 TAP'D.
.332 DIA.
.332 DRILL
1/2 HEX.


DWLA
SEC. 'A' WELL WARE LEAD




1.220
.900
1.535
1.650
7 HOLE DRILL
5 HEX
5/8-18
1
5
4
3
2

[PX 23, p 31]

NOTE:

MATERIAL: CARTRIDGE BRASS #42
.016 THICKNESS

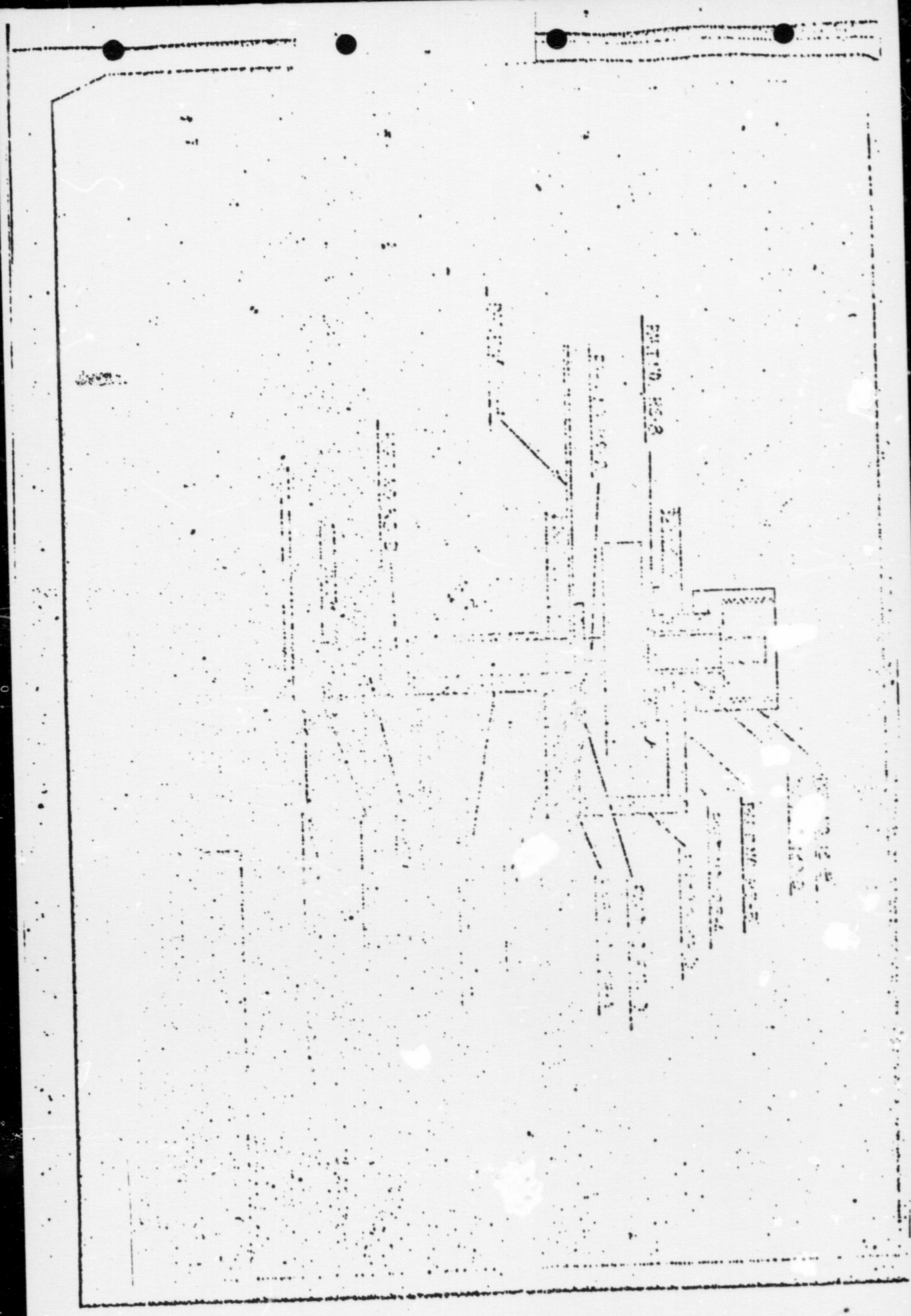
CONFIDENTIAL

	TITLE CONTACT HAT
SCALE 6X1	
MATERIAL	
DATE	

RS-68

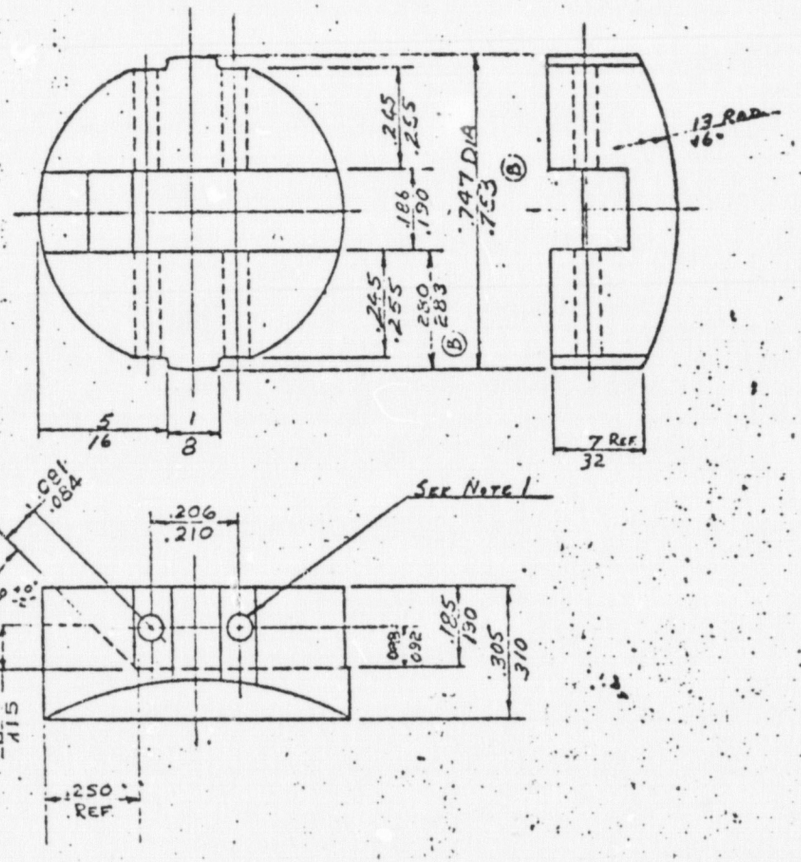
E-198

[PX 23, p 31]



[PX 23, p 32]

[PX 23, p 33]



NOTE:-

1. .063/.065 DRILL 2 HOLES IN LINE, AT 2 PLACES. .062/.0625 D.I.A. PIN MUST FIT THRU FREELY.
2. MAT.:- SINTALOY Z-802PB BRASS OR EQUAL-DENSITY APPROX 7.4 GMS./CC.
3. 125 FINISH ALL OVER.
4. .015 RAD. MAX. CORNERS & EDGES.
5. NICKEL PLATE .0003 MAX AFTER ASSEMBLY WITH 1- LINK

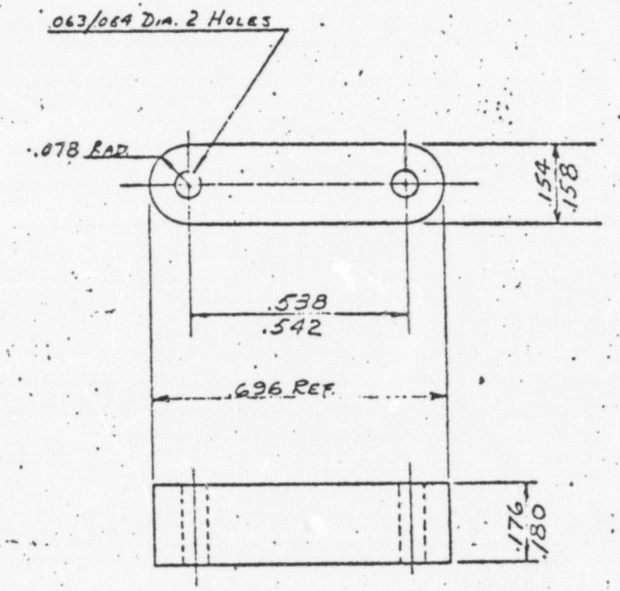
CONFIDENTIAL

							ANTI-SKID UNIT	TITLE GOVERNOR WEIGHT
							SCALE FULL	PERKIN ENGINEERING & DEVELOPMENT LTD
							MATERIAL AS OFF	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED			DRN BY / CHK BY / APP'D
1-27-53	B	[illegible]			BREAK ALL SHARP CORNERS AND REMOVE ALL BURR		PART	DRG NO 103-B
1-20-53	A	PLATING SPECIFICATION			FINISH OF EXCEPTIONAL ENGINEERING		FINISH	CHK BY / APP'D
DATE	LTR.	CHANGE RECORD	BY	CK.	DO NOT SCALE DIMENSIONS		HEAT 3	

[PX 23, p 33]

E-200

[PX 23, p 34]



NOTE:-

1. MATERIAL:- SINTALOY L-1004 IRON OR EQUAL
   DENSITY - APPROX. 6.2-6.4 GMS/CC
2. 125 FINISH ALL OVER.
3. .015 CHAMFER ALL EDGES.
4. NICKEL PLATE MAX. .0003 - ALL DIMENSIONS APPLY AFTER PLATING

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK
2-6-62	B	2 WAY 3 PLATE HOUSED		
2-	A			

DO NOT SCALE DIMENSIONS

ANTI-SND UNIT

TITLE GOVERNOR WEIGHT ASSY.

SCALE 4/1

MATL SEE NOTE

FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED

BREAK ALL SHARP EDGES AND RUN IT ALL OVER

DECIMALS OR FRACTIONS ± 1/64

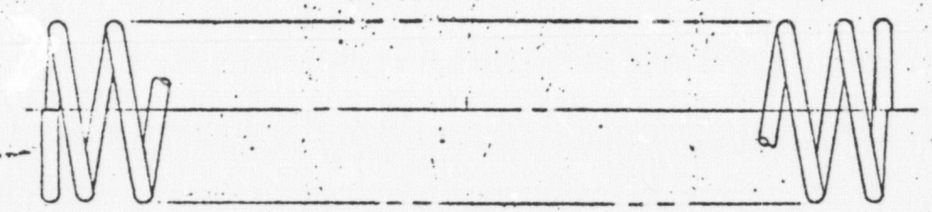
REQ'D 6

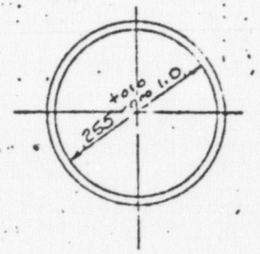
DRAWING NO. 104-13

[PX 23, p 34]

E-201

[PX 23, p 35]





DIAMETER OF WIRE ..... .025
ACTIVE COILS 14
TOTAL COILS ..... 16 APPROX.
DIRECTION OF HELIX ..... OPTIONAL
FREE LENGTH 1.380-1.395
LOAD AT COMPRESSED LENGTH
OF .750 IN. = 23 OZ. ± 5%

ENDS SQUARED, NOT GROUND
MATERIAL: .025 MUSIC WIRE
BURR SHARP EDGES

CONFIDENTIAL

						ANTI-SKID UNIT	TITLE GOVERNOR SPRING
						SCALE 5 X 1	PERRY DRILLING & DEVELOPMENT CO.
						MATL SEE NOTE	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DRAWN BY DATE
1-29-41	B	5.07 ADDED			BREAK ALL SHARP EDGES AND REMOVE ALL BURRS	PATT	CHK'D BY DATE
8-16-41	A	WIRE SIZE WAS .024			ALL DIMENSIONS ARE FOR FREE CONDITION	FINISH	ENG'R BY DATE
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQD 1	APPROV'D BY DATE JJ

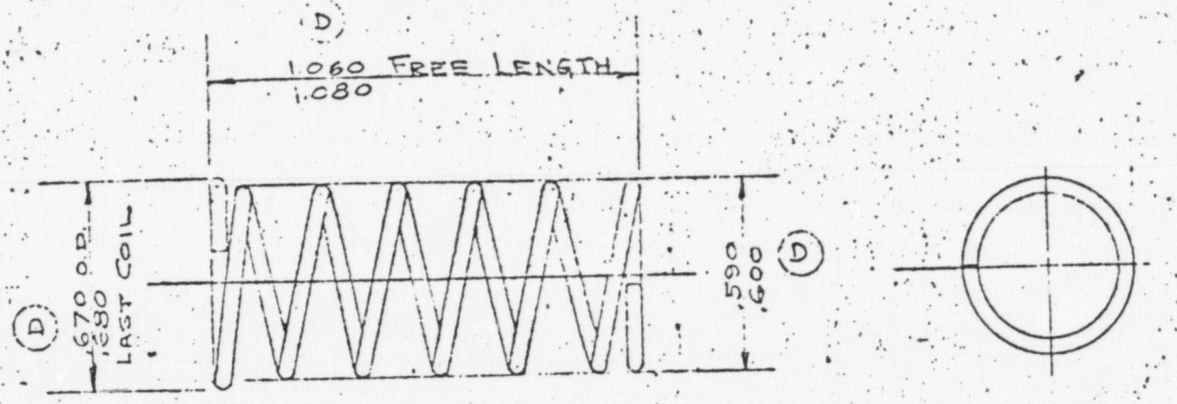
105-B

[PX 23, p 35]

E-202

[PX 23, p 36]

#24



DIAMETER OF WIRE _____ .038
NOMINAL O.D. _____ .590-.600
TOTAL ACTIVE COILS _____ 7½
DIRECTION OF HELIX _____ OPTIONAL
FREE LENGTH _____ 1.040-1.060
LOAD AT COMPRESSED LENGTH OF 9/16 IN = 20.02 ± 5%

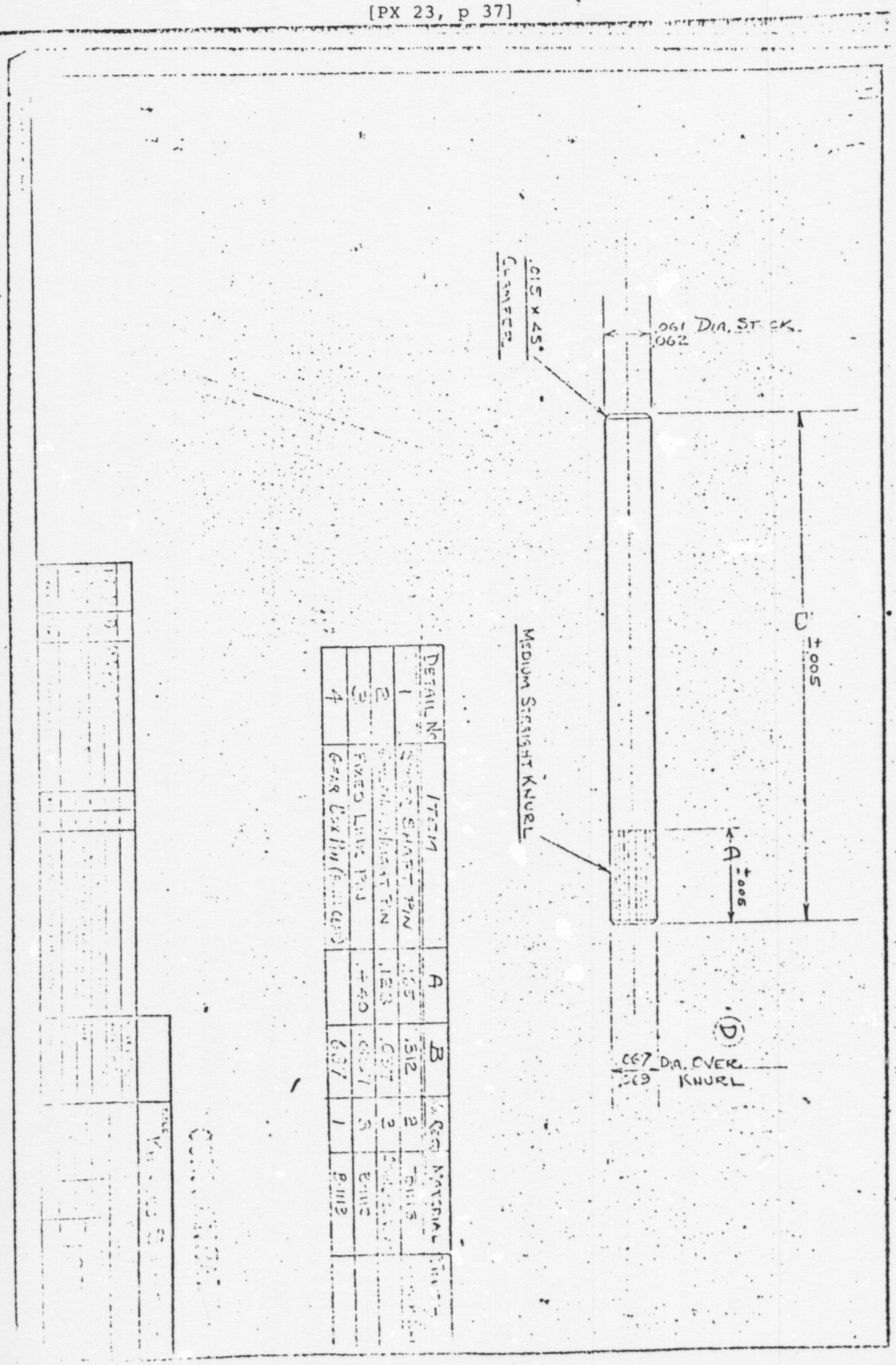
ENDS SQUARED, NOT GROUND
MATERIAL: MUSIC WIRE
BURR SHARP EDGES

CONFIDENTIAL

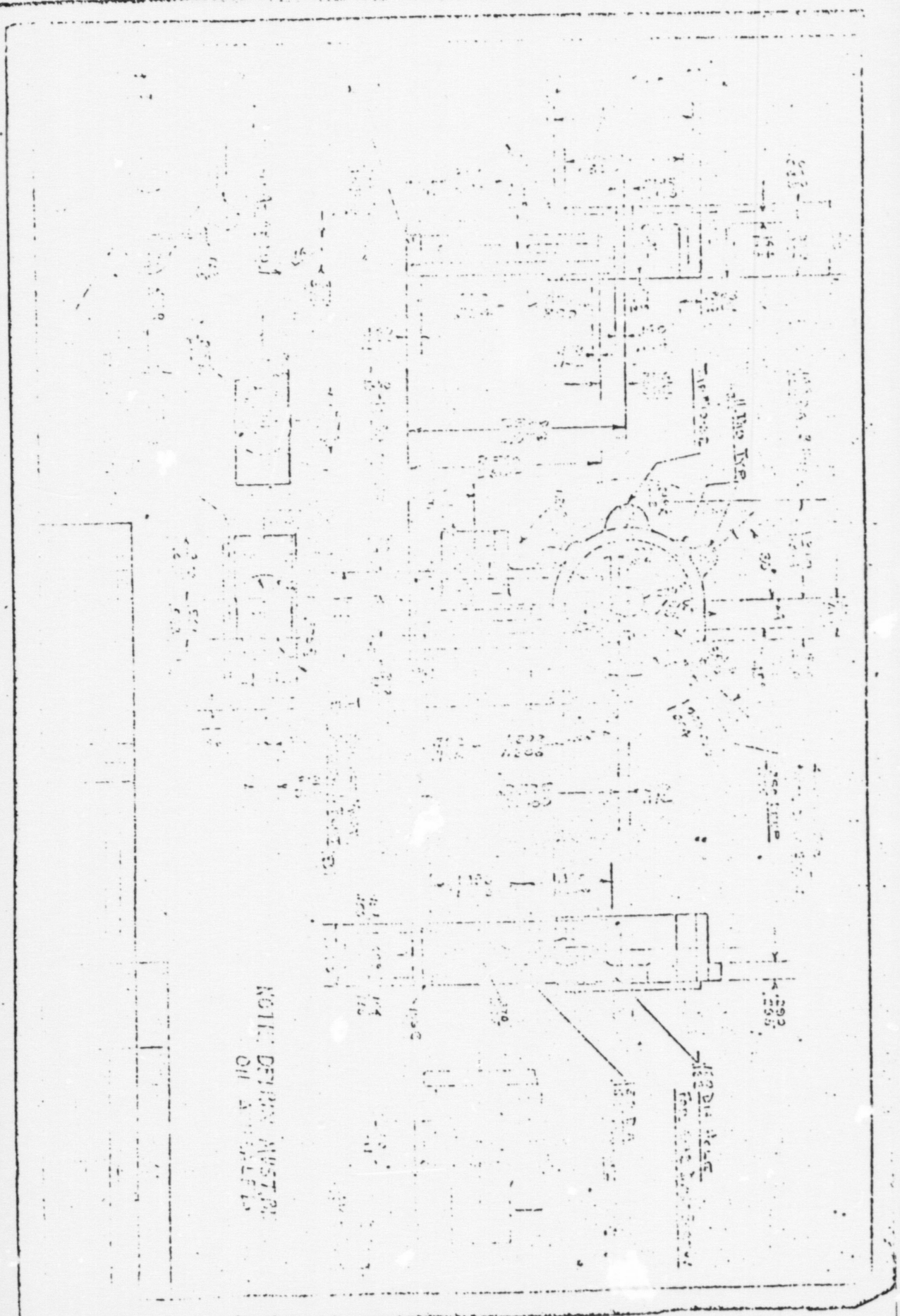
12-61	A	DESIGN CHANGED				ANTI-SKID UNIT	SPRING, ARMATURE
8-26-6	B	WIRE DIA & LOAD CHANGED				SCALE 3 X 1	
						MATL SEE	
						NOTE	
						PATT	
						CUSTOM	106-E
DATE	STD	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D 1	

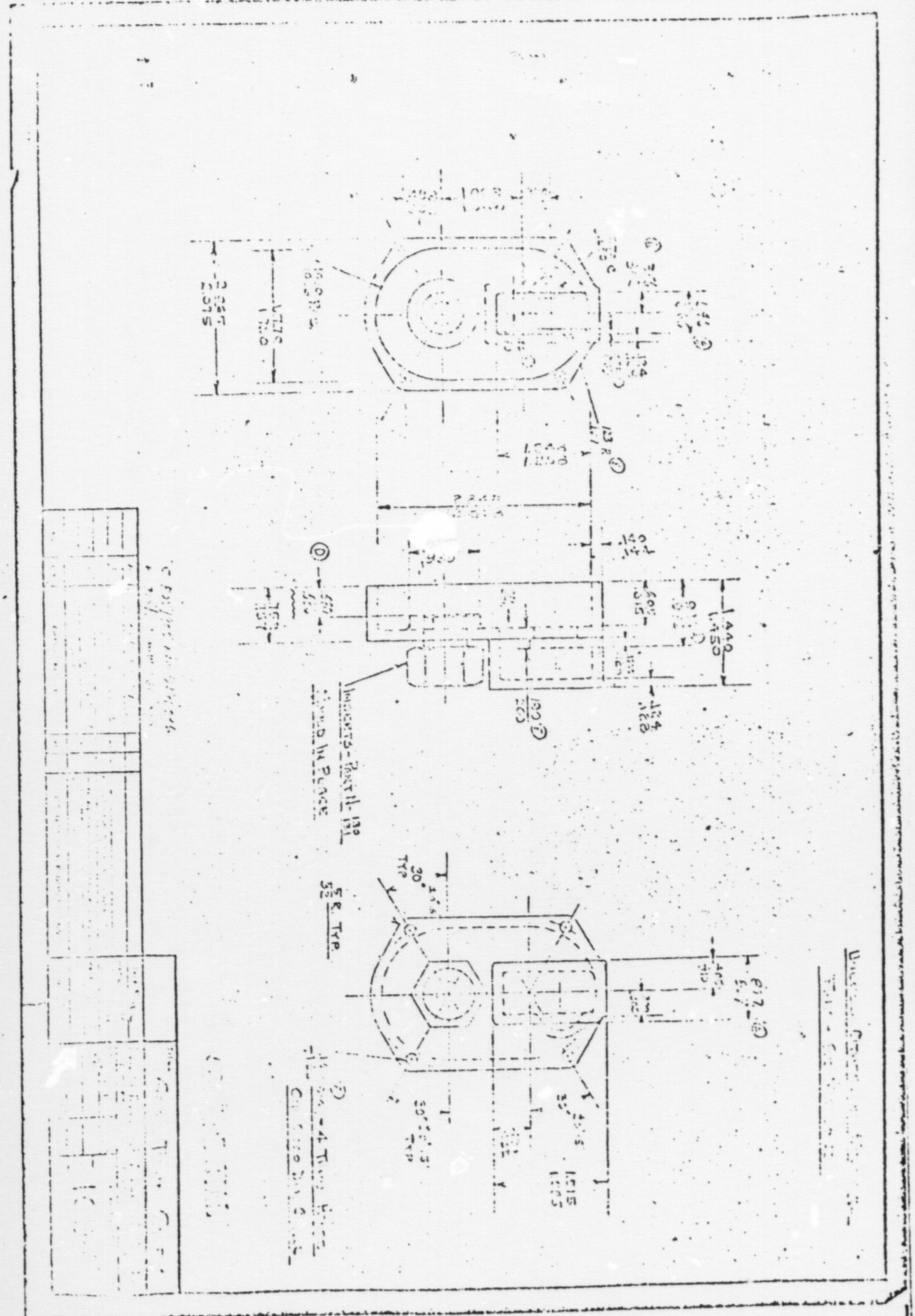
[PX 23, p 36]

E-203



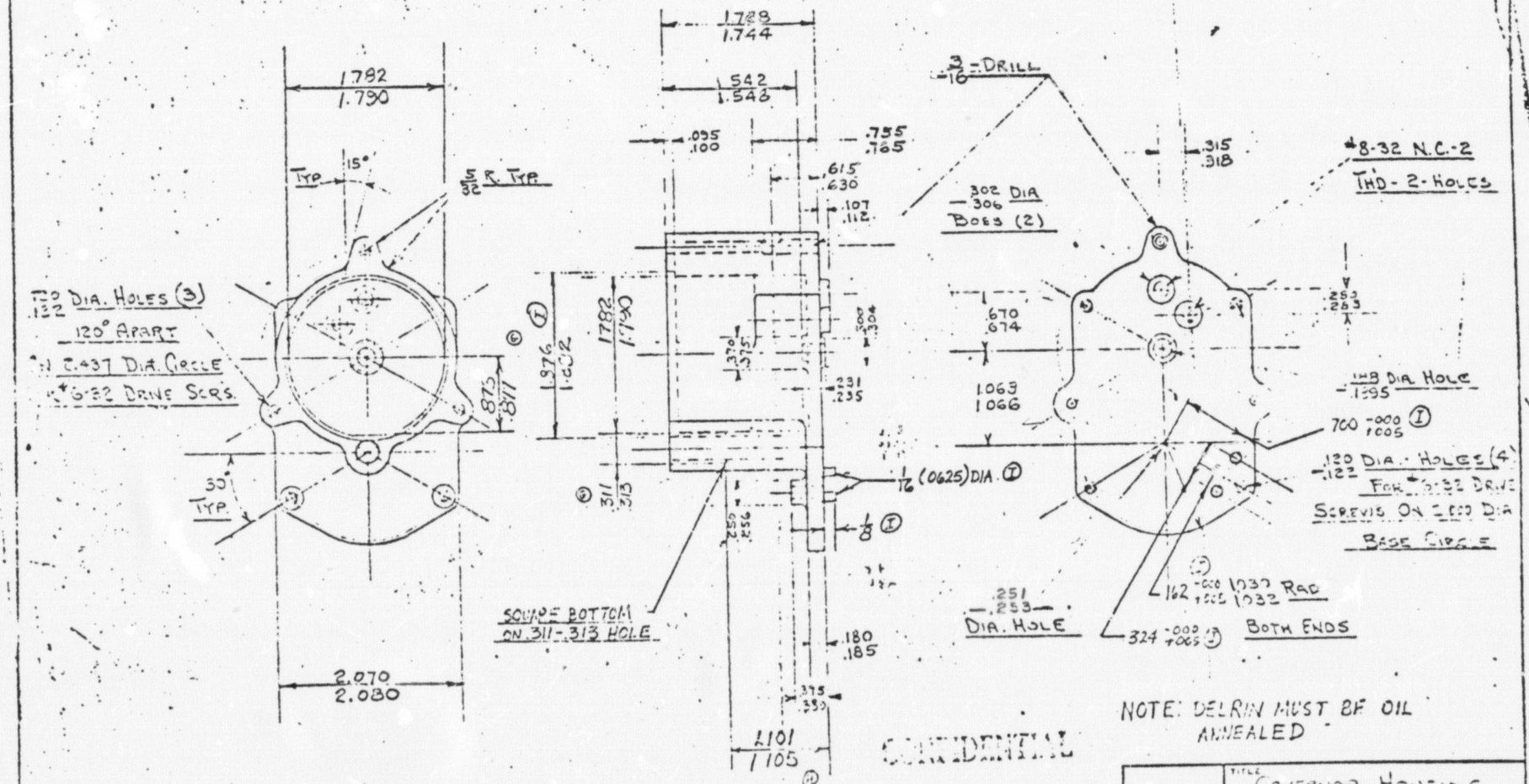
DETAIL NO	ITEM	A	B	REQ	MATERIAL
1	SHAFT SPACER PIN	.125	.312	2	TOOL STEEL
2	SHAFT PIN	.125	.687	2	TOOL STEEL
3	FIXED LINK PIN	.250	.687	2	TOOL STEEL
4	GEAR BOX PIN (SHORT)		.687	1	BRASS





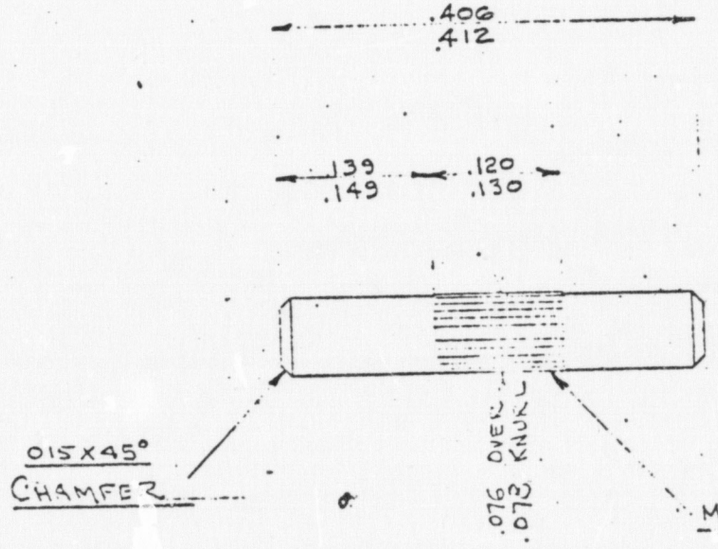
[PX 23, p 39]

[PX 23, P 40]


1.782
1.790
TYP
15°
5/32 R. TYP
.170
.172 DIA. HOLES (3)
120° APART
ON C.437 DIA. CIRCLE
FOR 6-32 DRIVE SCRS.
30°
TYP.
.875
.811
1.788
1.744
1.542
1.548
.095
.100
.755
.765
.615
.630
.107
.112
1.782
1.790
1.876
1.602
.370
.375
.304
.300
.231
.235
.311
.313
.259
.256
SQUIRE BOTTOM
ON .311-.313 HOLE
.180
.185
.375
.350
1.101
1.105
3/16 DRILL
.315
.318
8-32 N.C-2
THD-2-HOLES
.302 DIA
.306
BOSS (2)
.670
.674
1.063
1.066
.253
.253
.188 DIA HOLE
.1895
.700 -.000 +.005
.130 DIA. HOLES (4)
.122
FOR 6-32 DRIVE
SCREWS ON 1.500 DIA
BASE CIRCLE
1/16 (.0625) DIA
1/8
.251
.253
DIA. HOLE
.162 -.000 +.005
1.037
1.032 RAD
BOTH ENDS
.324 -.000 +.005
NOTE: DELRIN MUST BE OIL ANNEALED
CONFIDENTIAL

| | | |

[PX 23, p 41]



NOTE

HARDEN & DRAW TO

C-45-47 ROCKWELL

NICKEL PLATE .0003 MIN.

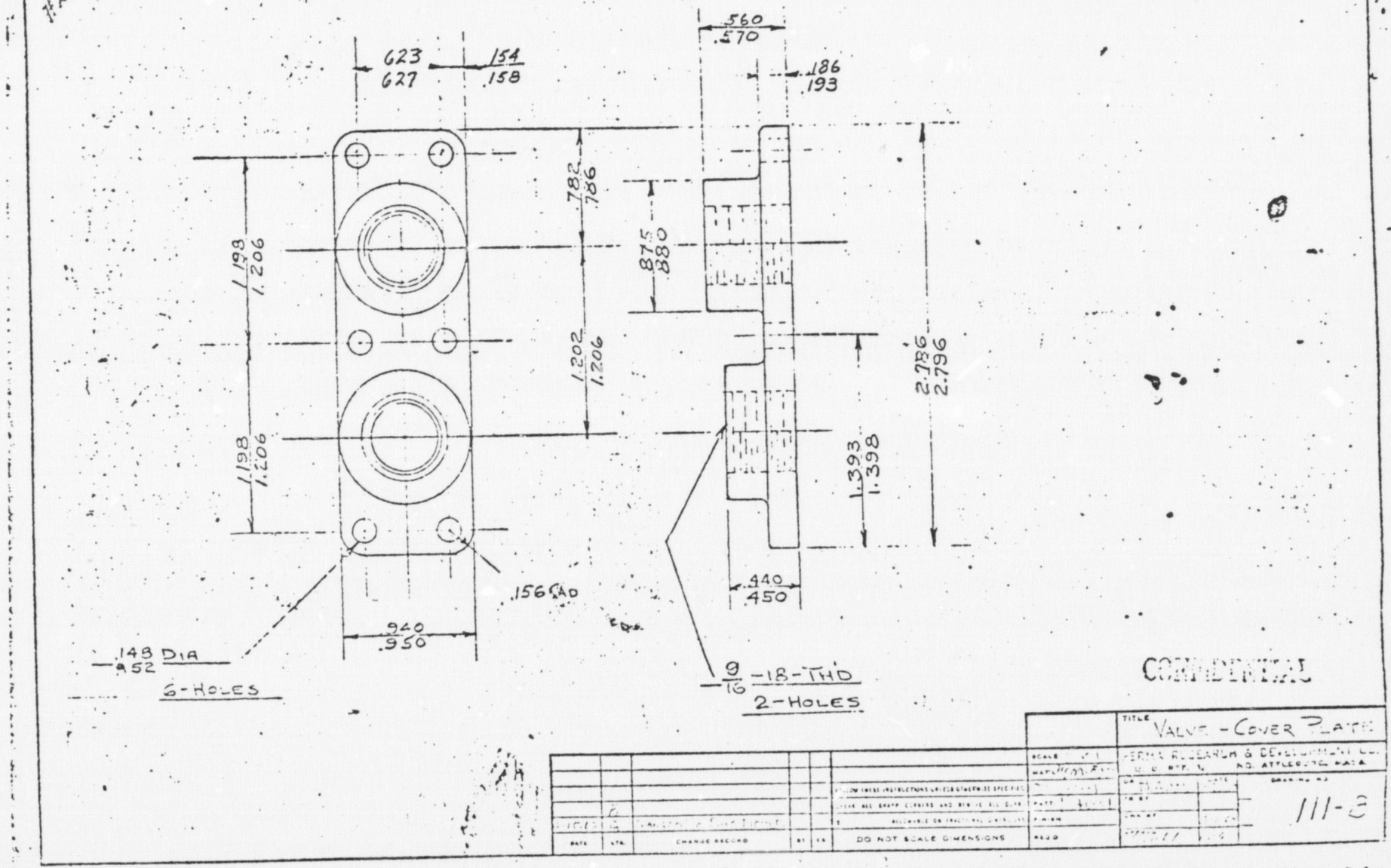
					SCALE 10 X 1	TITLE CAM GEAR SHAFT PIN
					MAT'L 10G 2 —	PERMA RESEARCH & DEVELOPMENT CO.
				UNLESS THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	TOL. .XX ±.02	DRAWN JWW 3-2-62
				BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	CHK'D
12-1-62	A	CHANGE TO NICKEL		X ALLOWANCE ON FRACTIONAL DIMENSIONS		APPROV'D
DATE	STR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REV'D

DRAWING NO. 113A

E-208

[PX 23, p 41]

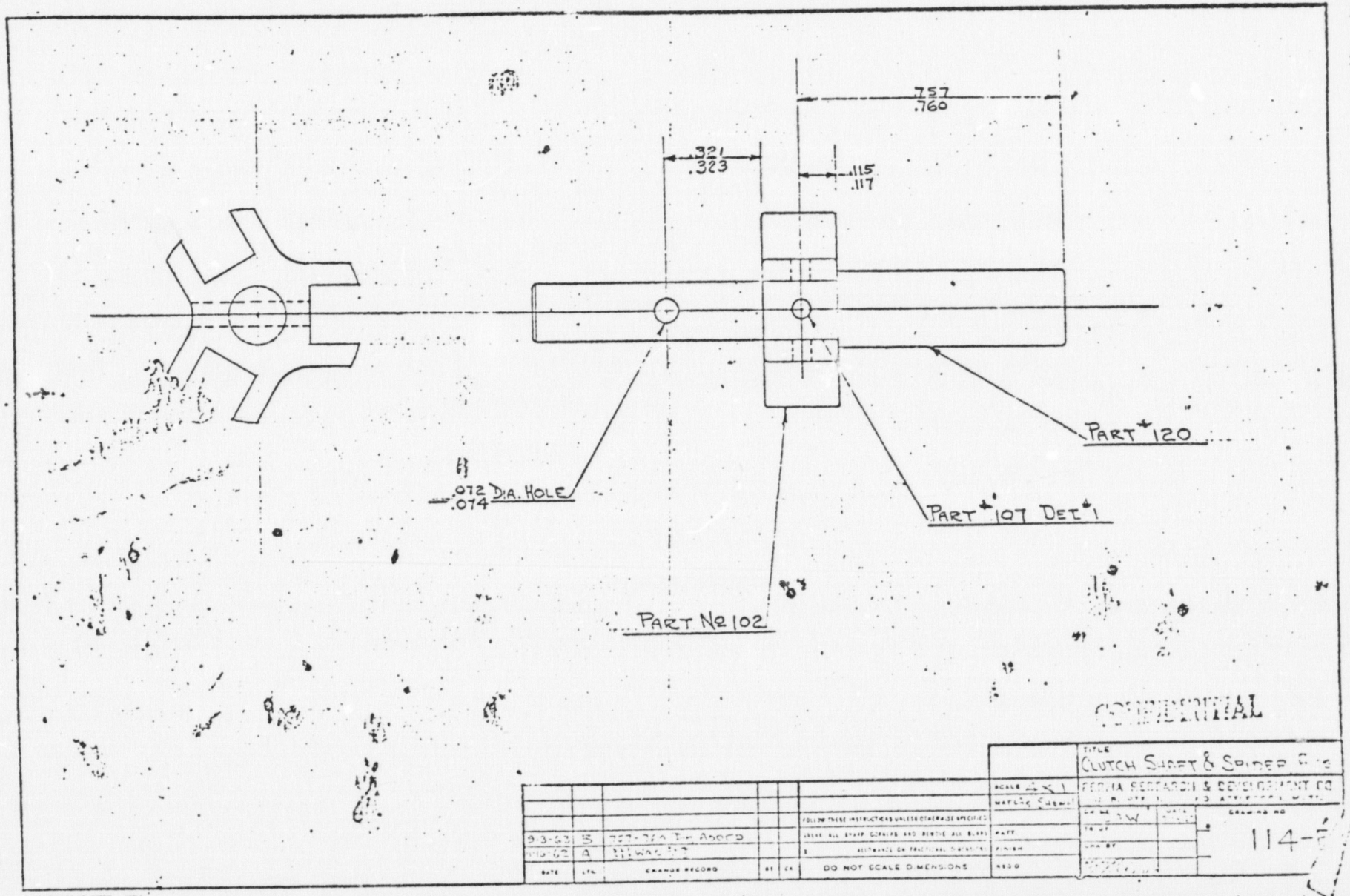
[PX 23, p 42]



[PX 23, p 42]

E-209

[PX 23, p 43]



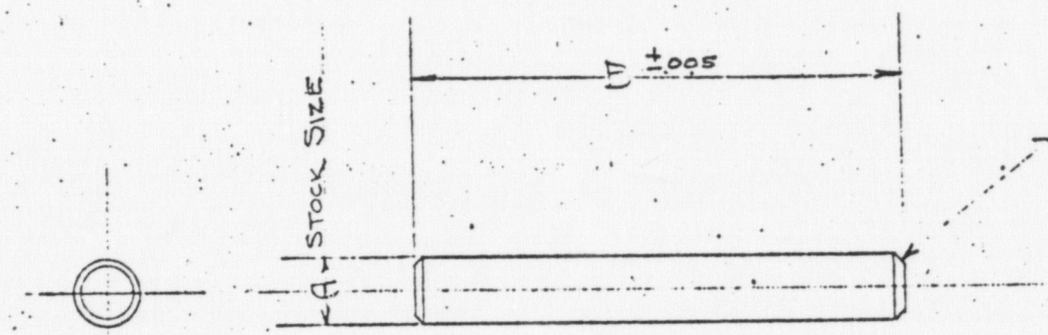
CONFIDENTIAL

					SCALE 4 X 1	TITLE CLUTCH SHAFT & SPIDER
					MATERIAL SUBMIT	FERMA RESEARCH & DEVELOPMENT CO
				FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DR. BY JW DATE
9-3-63	5	757-760 I.D. ADDED		BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PATT.	CH. BY
9-10-62	A	.111 WAS .112		X ALLOWANCE OR TOLERANCE ON FRACTIONAL DIMENSIONS	FINISH	APP. BY
DATE	LTR.	CHANGE RECORD	BY	CK. DO NOT SCALE DIMENSIONS	HEAT TR.	DRAWING NO. 114-[illegible]

[PX 23, p 43]

E-210

[PX 23, p 44]



DETAIL NO	NAME	A	B	FINISH	# REQ'D	MAT.
1	TRANSMISSION SHAFT PIN	.078	.562	AS DRAWN	1	STAINLESS STEEL
2	SPIDER LINK PIN	.0625	.468	AS DRAWN	6	STAINLESS STEEL

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK
4-13-62	A	MAT. CHANGE ON SPIDER PIN		

DO NOT SCALE DIMENSIONS

SCALE 2X1

TITLE PINS

DRAWN BY [illegible]

CHK BY

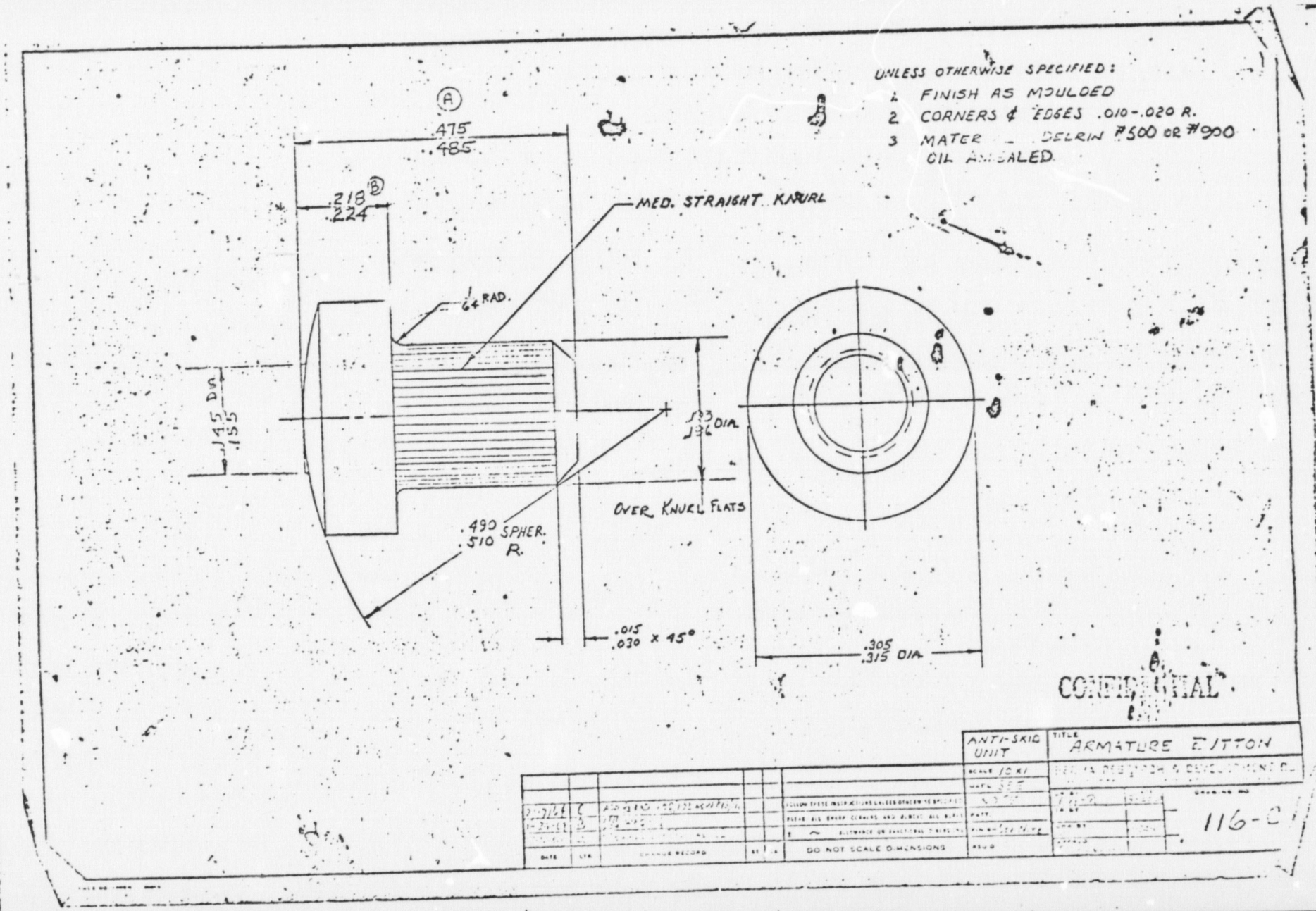
APPD BY

DRAWING NO 113-A

[PX 23, p 44]

E-211

[PX 23, p 45]



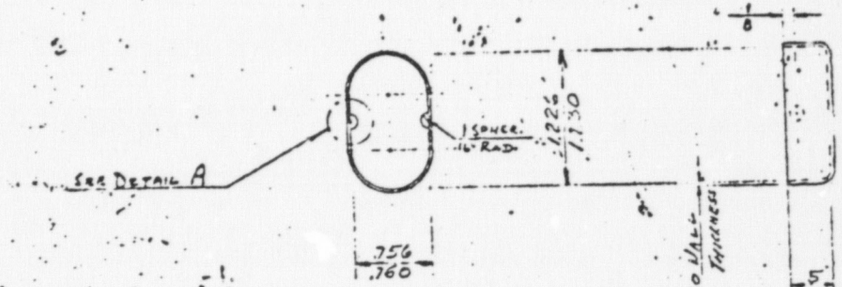
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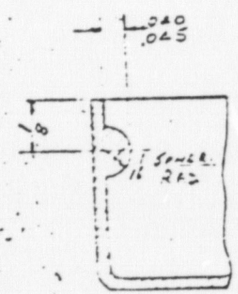
E-212

[PX 23, p 46]

[PX 23, p 46]

E-213





DETAIL "A" MAT'L PLASTIC
SCALE 4:1 DELRIN

DATE	LTR	CHANGE RECORD	BY	CK

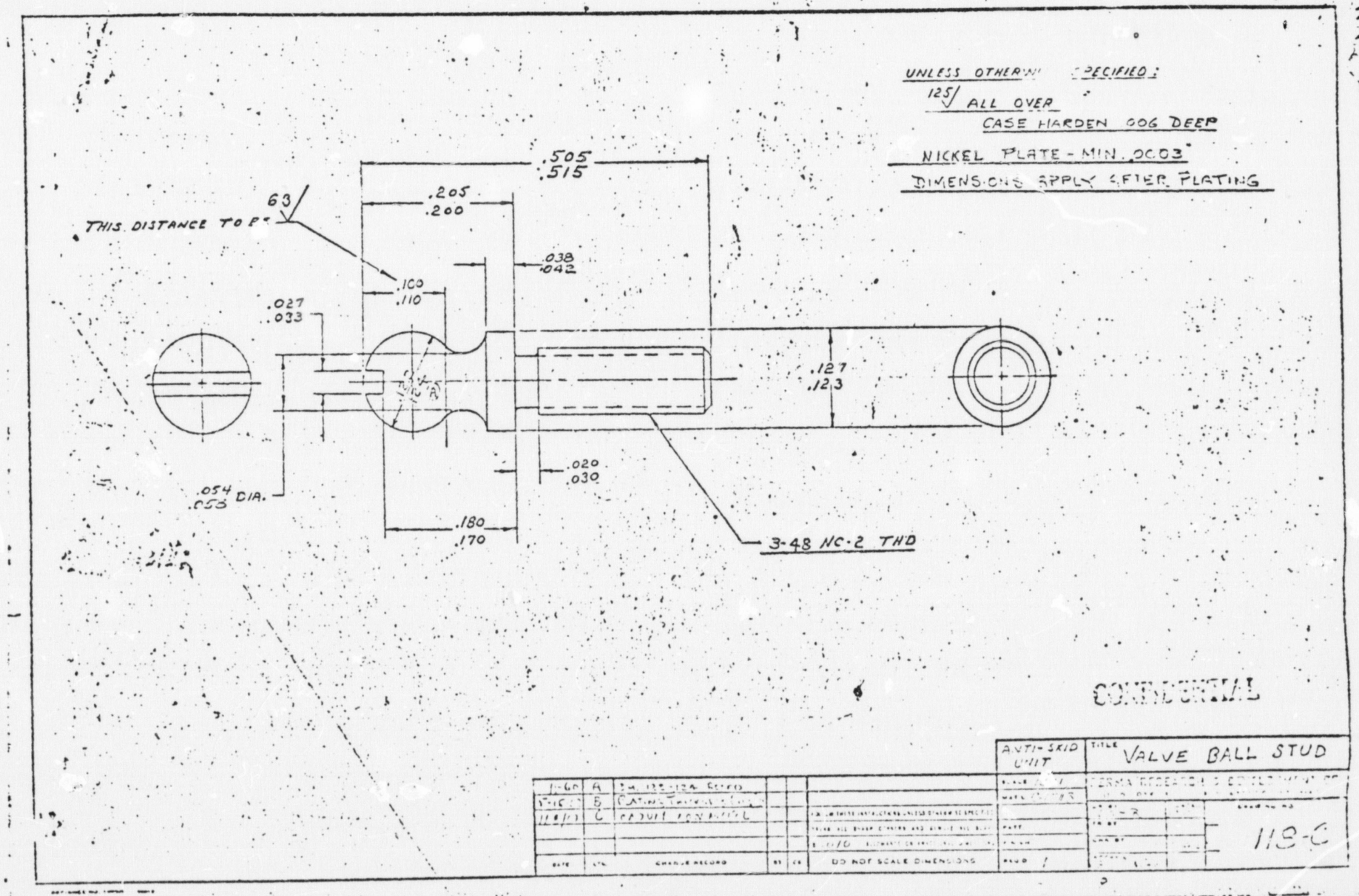
DO NOT SCALE DIMENSIONS

ANTI SKID UNIT

TITLE VALVE HOUSING COVER

115

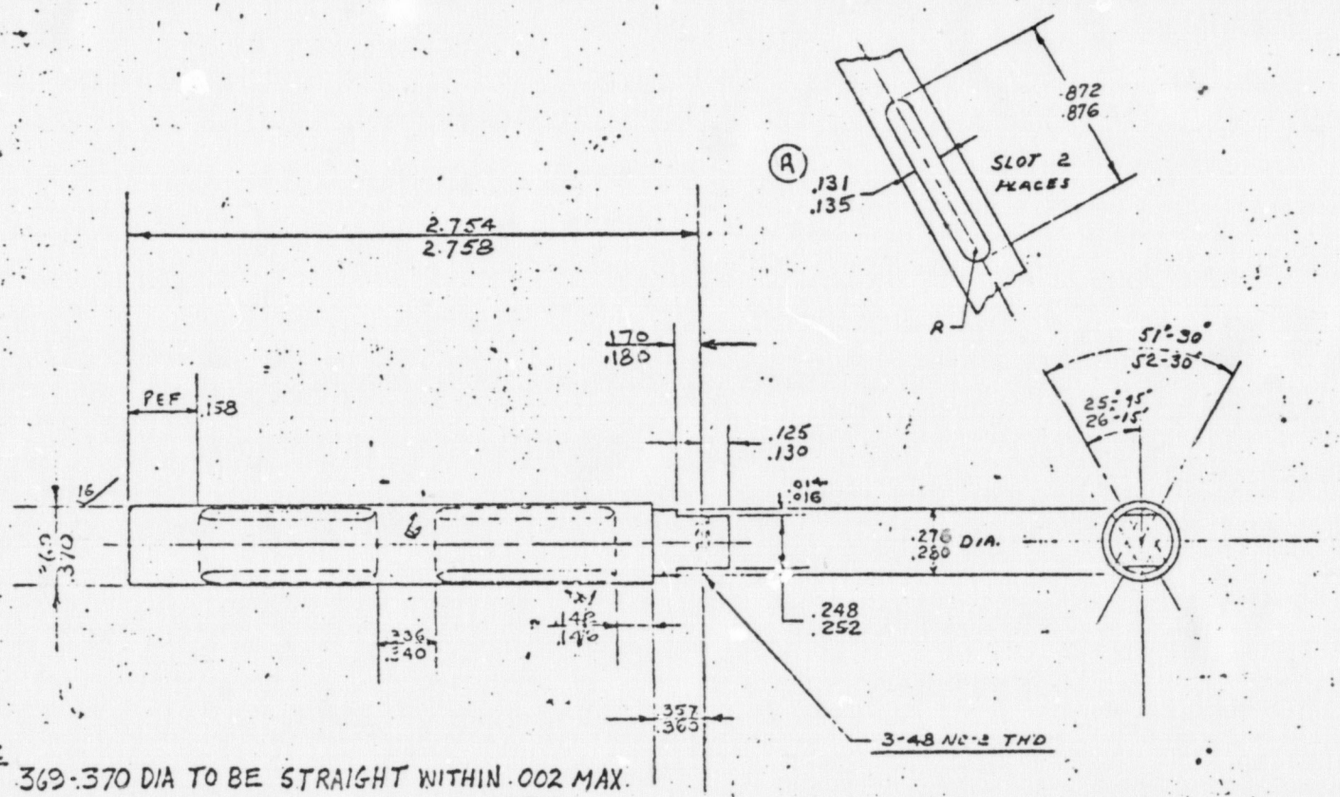
[PX 23, p 47]



[PX 23, p 47]

E-214

[PX 23, p 48]



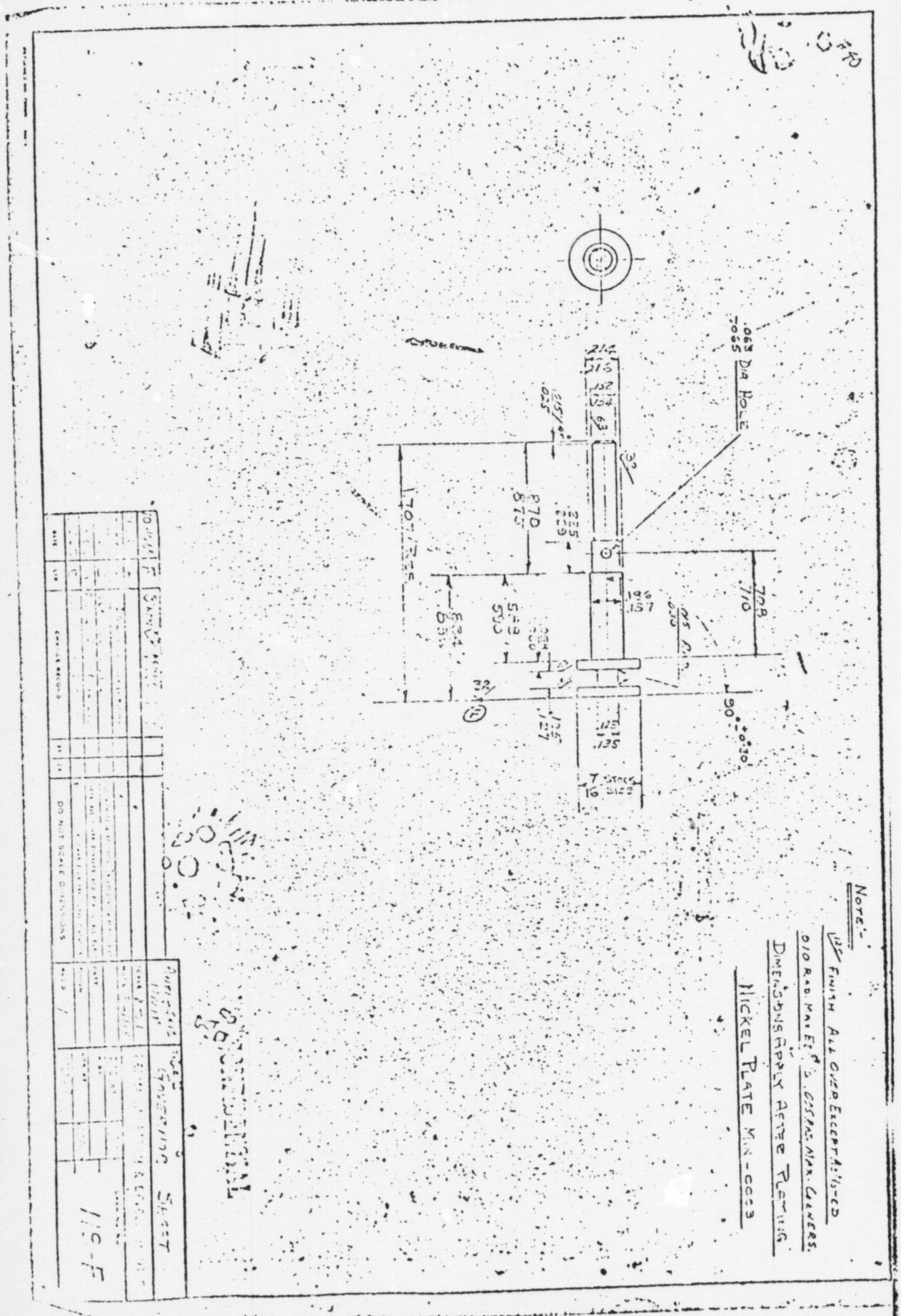
NOTE:
369-.370 DIA TO BE STRAIGHT WITHIN .002 MAX.
PORTS TO BE BURR AND FLASH FREE

CONFIDENTIAL

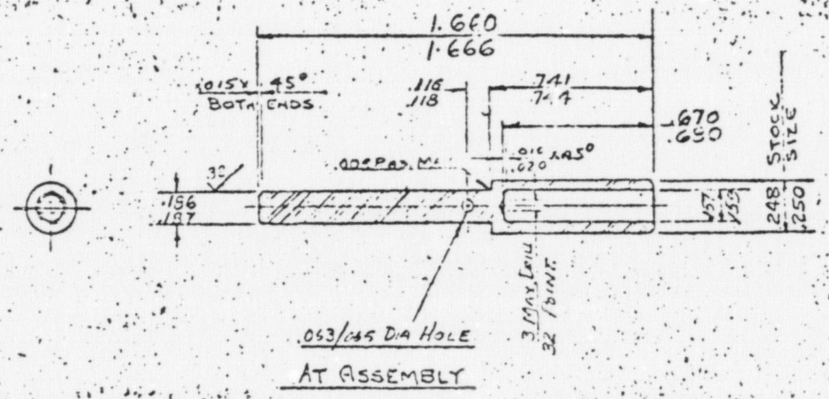
	A	.132 WAS .125				ANTI-SKID UNIT	TITLE VALVE
	C	LENGTHS CHANGED				SCALE 2X1	PERMA RESEARCH & DEVELOPMENT CO.
							DRAW NO. 117-E
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS		

E-215

[PX 23, p 48]


.063 DIA HOLE
.055
.070
.875
.710
.500
.135
90°
Notes:
Finish All Over Except As Noted
010 Rad. Max. Ex. Unless Max. Clearers.
Dimensions Apply After Plating
Nickel Plate Min. .0003
Do Not Scale Dimensions

[PX 23, p 50]



Note:

.010 RAD. MAX. EDGES

32 FINISH ALL OVER.

NICKEL PLATE - MAX. .0003

DIMENSIONS APPLY AFTER PLATING

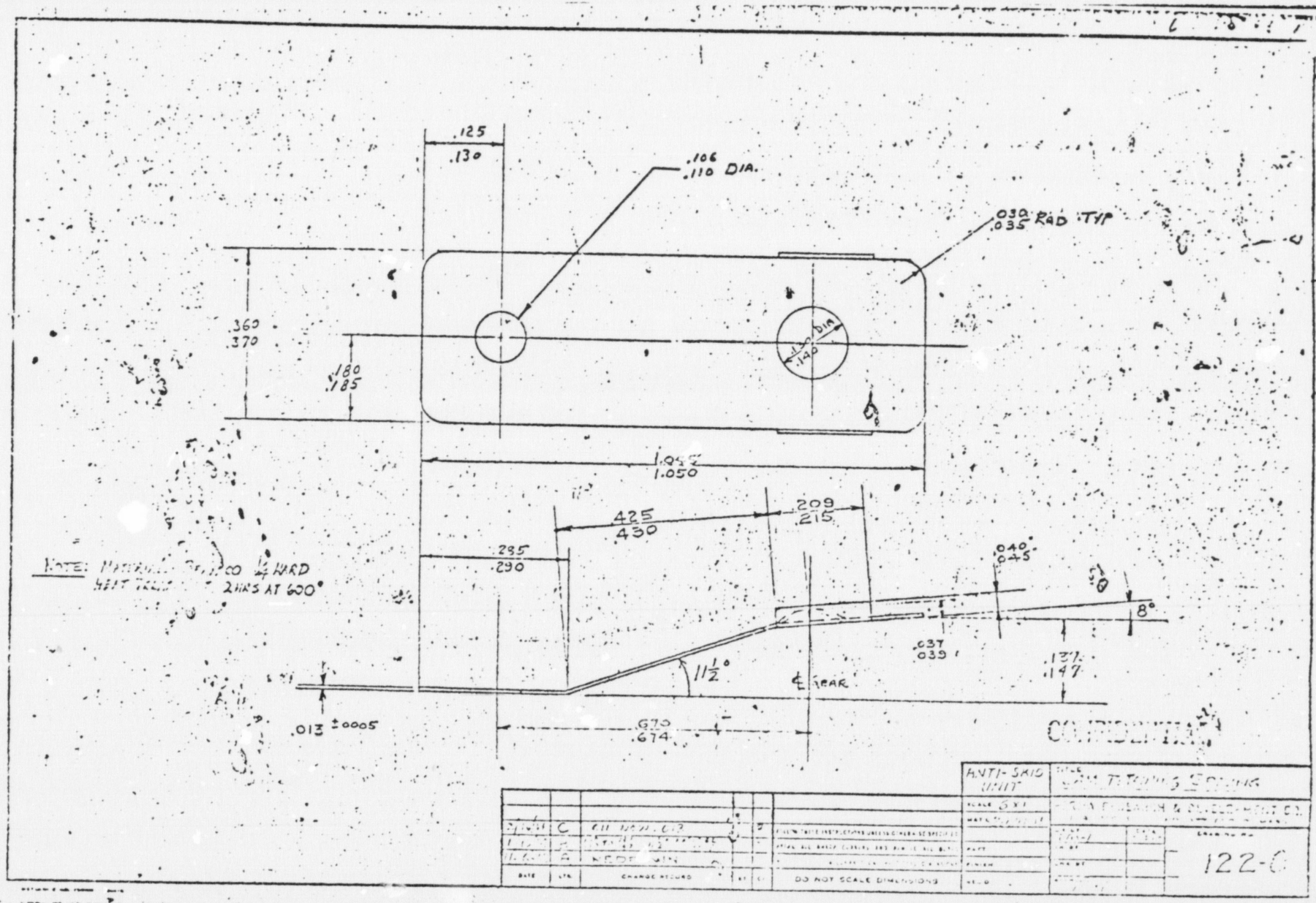
CONFIDENTIAL

						ANTI-SKID UNIT	TITLE GOVERNOR CLUTCH SHAFT
						SCALE 2/1	
						MATL. 1144	
12363	C	PLATING CHANGE					
							DRAWING NO. 120-C
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	FINISH 1	

[PX 23, p 50]

E-217

[PX 23, p 51]

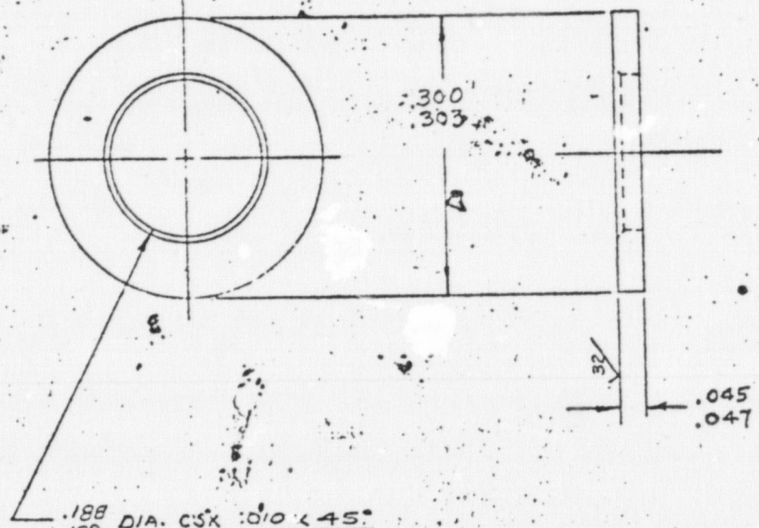


[PX 23, p 51]

E-218

[PX 23, p 52]

UNLESS OTHERWISE SPECIFIED:
125/ ALL OVER
EDGES .015 R. MAX.
NICKEL PLATE .0001 MIN
CASE HARDEN .006-.008 DEEP
DIMENSIONS APPLY AFTER PLATING



CONCENTRIC WITH O.D. - .004 T.I.R.

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK

DO NOT SCALE DIMENSIONS

ANTI-SKID UNIT

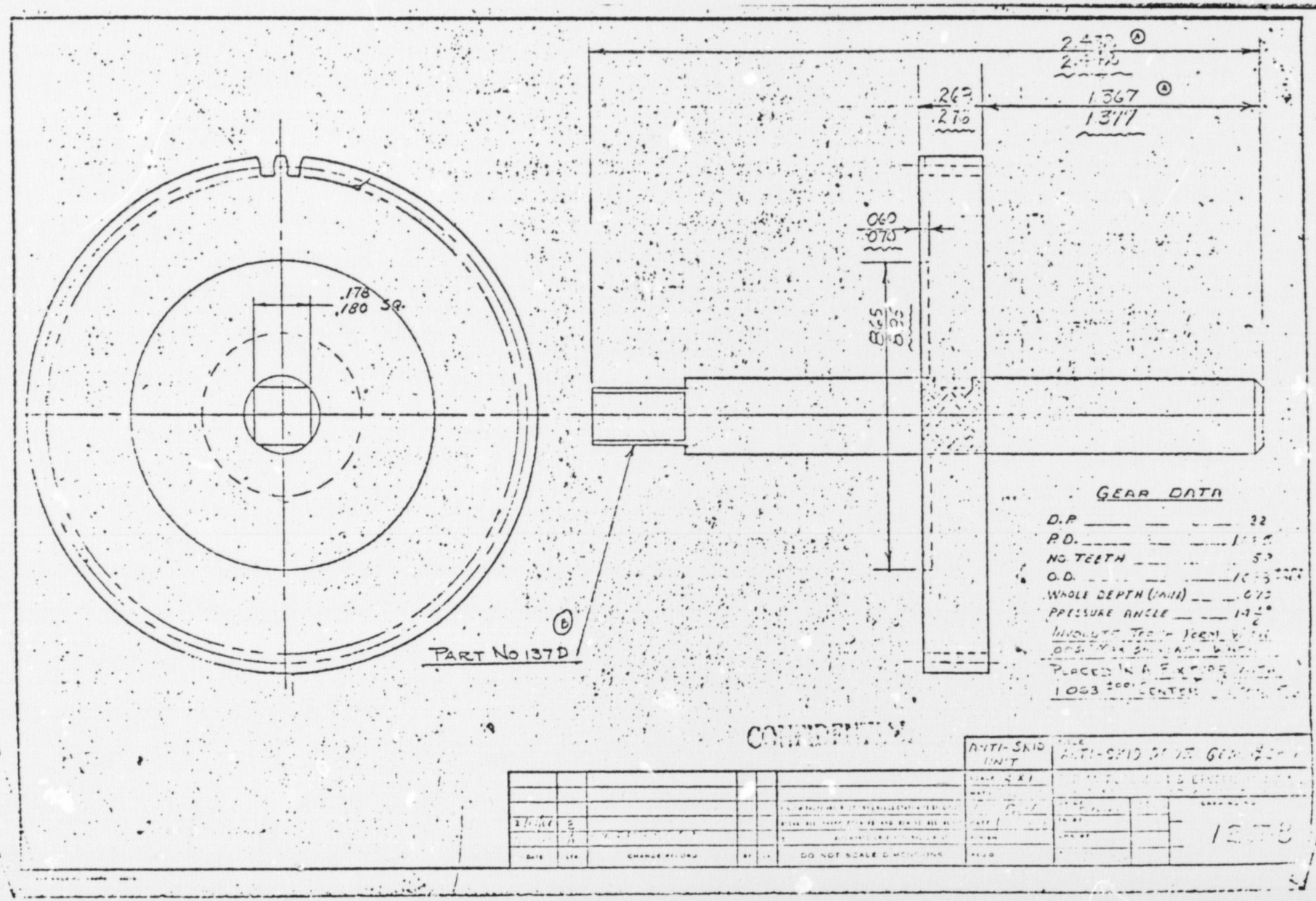
TITLE Gov SHAFT THRUST WASHER

FERRIS RESEARCH & DEVELOPMENT CO.

121-A

[PX 23, p 52]

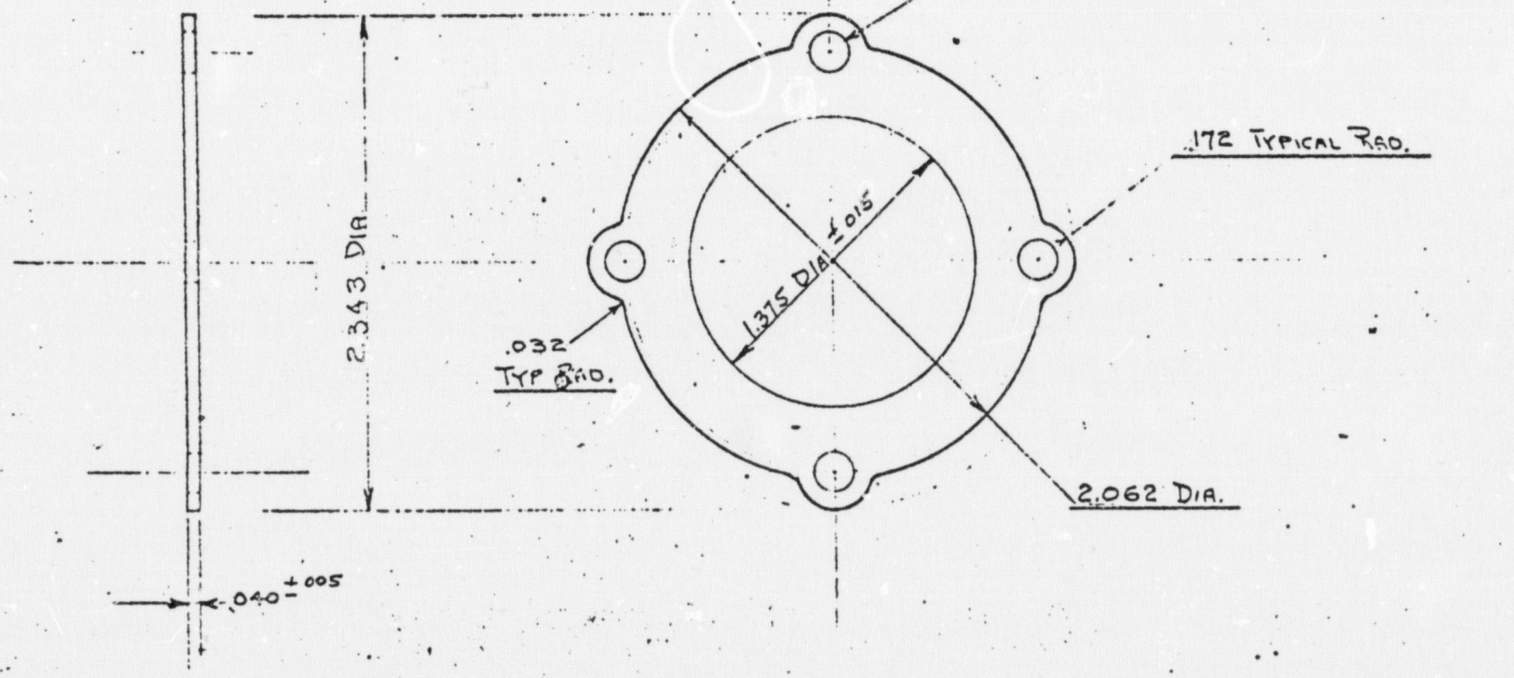
[PX 23, p 53]



[PX 23, p 53]

E-220

[PX 23, p 54]



CONFIDENTIAL

						SCALE 2 X 1	TITLE SOLENOID GASKET
						MATERIAL	PERMA RESEARCH & DEVELOPMENT CO.
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	FINISH	DRAWN
11-9-62	B	CHANGED OR. TOLERANCE			BREAK ALL SHARP CORNERS AND REMOVE ALL BURR.	PATT	CHK'D
11-7-62	A	TOLERANCE CHANGES			± .005 DECIMALS ± 1/64 FRACTIONAL UNLESS SPEC.	FINISH	APP'D
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D	DWG. NO. 126-B

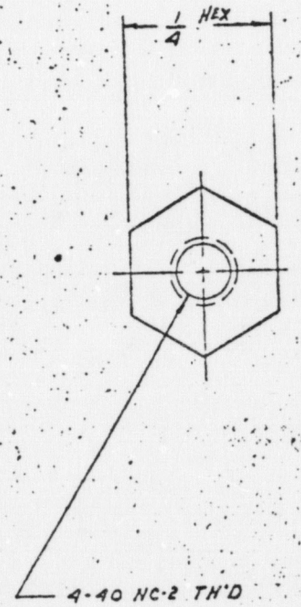
[PX 23, p 54]
E-221

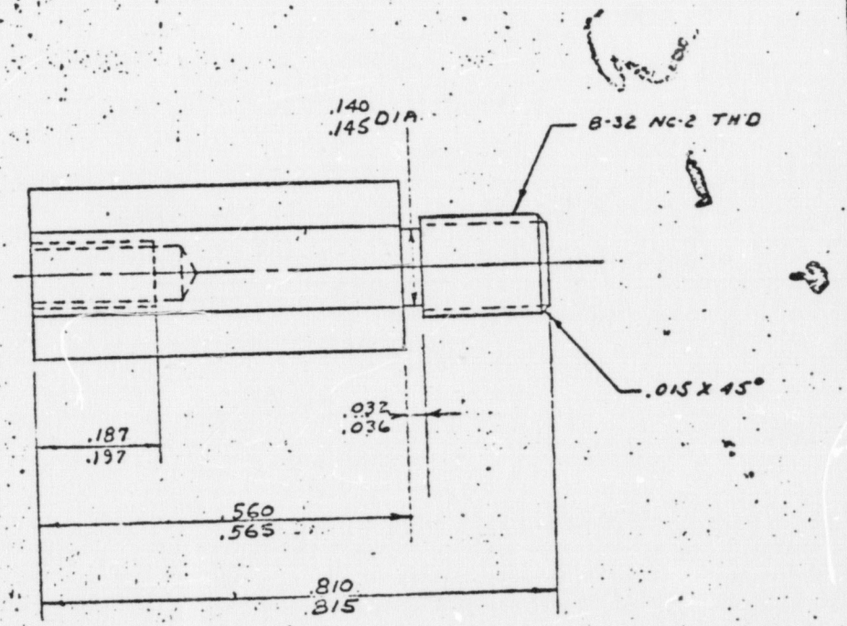
[PX 23, p 55]

[PX 23, p 55]

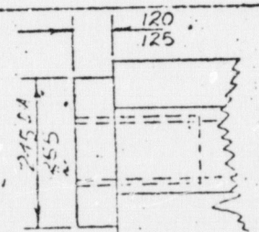
E-222

UNLESS OTHERWISE SPECIFIED:
125√ ALL OVER
EDGES .010-.015 R





DETAIL 2



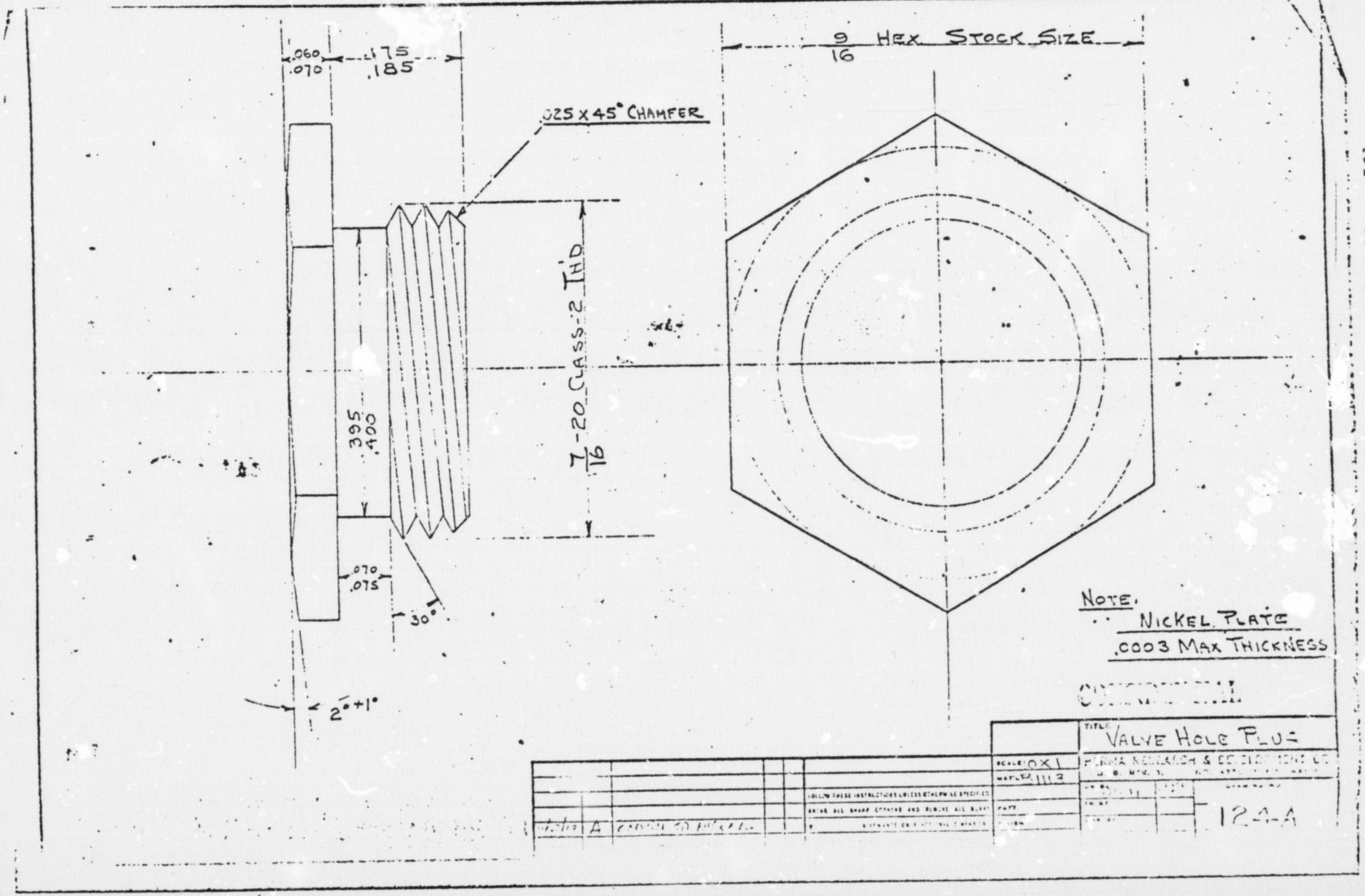
NOTE:
THIS POST IS USED UNDER MICRO SWITCH MOUNTING BRACKET

CONFIDENTIAL

				ANTI-SKID UNIT	TITLE CAM RETAINING SPRING POST
				FINISH C.P.I.	BENDIX RESEARCH & DEVELOPMENT DIV.
				MATERIAL SAE 1020	
				FOR DETAILS OMIT 2	
				AFTER ALL MACH. OPERS. AND BRAZE ALL PTS.	DATE
				2 C.I.D. ALL DRAWINGS	
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS
		DETAIL 2			

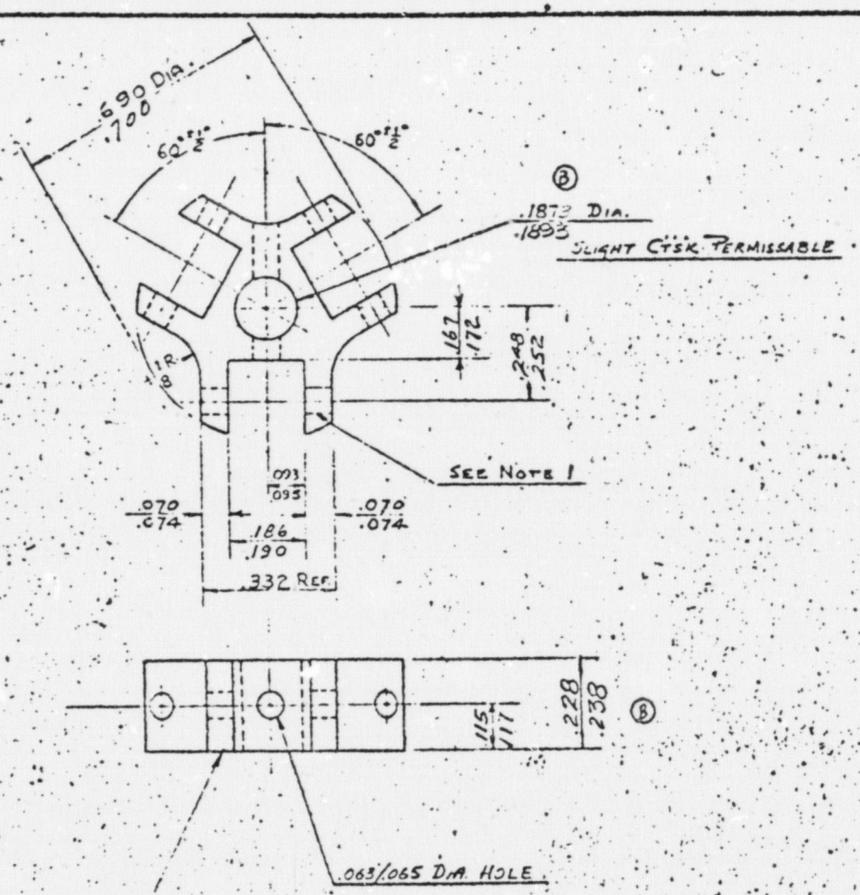
SIZE A | DWG. NO. 1234

[PX 23, p 56]



[PX 23, p 56]

[PX 23, p 57]



NOTE:-

1. .0635/.066 DRILL 2 HOLES IN LINE AT THREE PLACES. — .062/.0625 DIA. PIN MUST FIT THRU FREELY.
2. √125 FINISH ALL OVER.
3. .015 RAD. MAX. EDGES & CORNERS.
4. MAT: SINTALOY L1-1010 INFILTRATED IRON DENSITY 7.2 - 7.4 Gms./CC.
6. FLASH NICKEL PLATE

CONFIDENTIAL

						AUTH. SKD UNIT	TITLE SPIDER
DRAWN							DRAWING NO. 102-3
DATE		CHARLIE ALLARD			DO NOT SCALE DIMENSIONS	SHEET 2	

[PX 23, p 57]

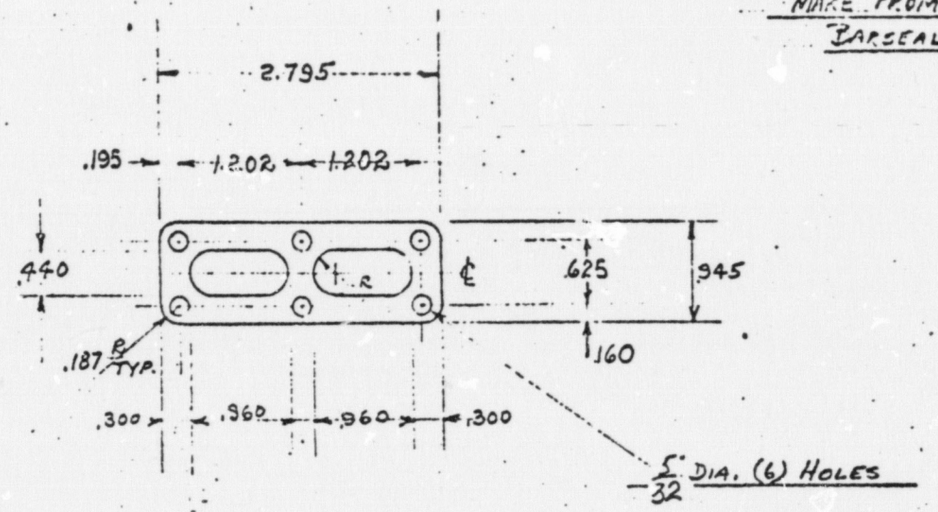
E-224

[PX 23, p 58]

[PX 23, p 58]

E-225

Note ±.005
MAKE FROM .040 THICKNESS
BARSEAL - A 8209

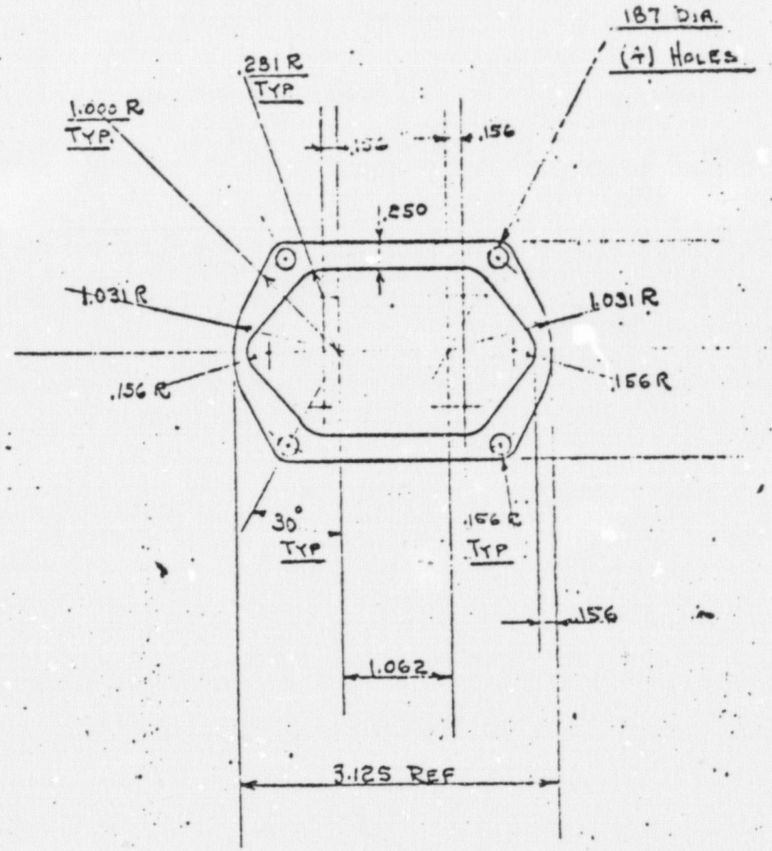


CONFIDENTIAL

						SCALE FULL	TITLE Valve Cover Plate Gasket
						MATERIAL PLATE	PERKIN-ELMER & DEVELOPMENT CO.
							DRAWN
1-27-65	B	DIM CHANGED				DATE	TR BY
1-25-65	A	Tolerance Changed				FINISH	CHK BY
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MFG D	APPR BY

DRAWING NO 127-B

[PX 23, p 59]



NOTE:
MAKE FROM .040 ±.005 THICK
BARSEAL AISI 309
OR EQUIVILENT

CLASSIFIED

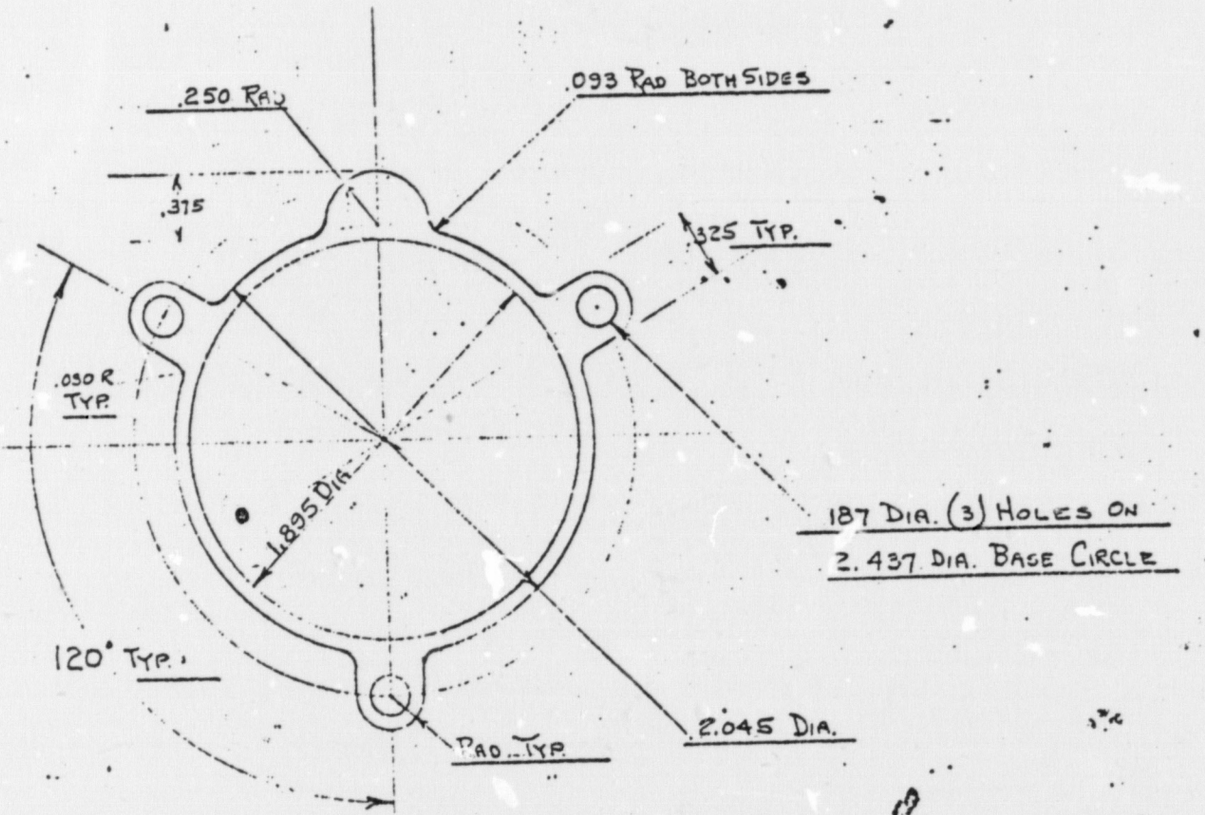
					SCALE FULL	TITLE GEAR TRAIN COLD SHIELD
					MATERIAL SEE NOTE	
				FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		
WAS 3	2	PARAGRAPH 4 ADDED		REMOVE ALL SHARP CORNERS AND BURRS	PATT	
7-14-67	A	THICKNESS CHANGED		± .010 ALLOWANCE ON TRIGONOMETRIC DIMENSIONS	FINISH	
DATE	LTR	CHANGE RECORD	BY	DO NOT SCALE DIMENSIONS	HEAT	

E- 426

[PX 23, p 59]

[PX 23, p 60]

[PX 23, p 60]

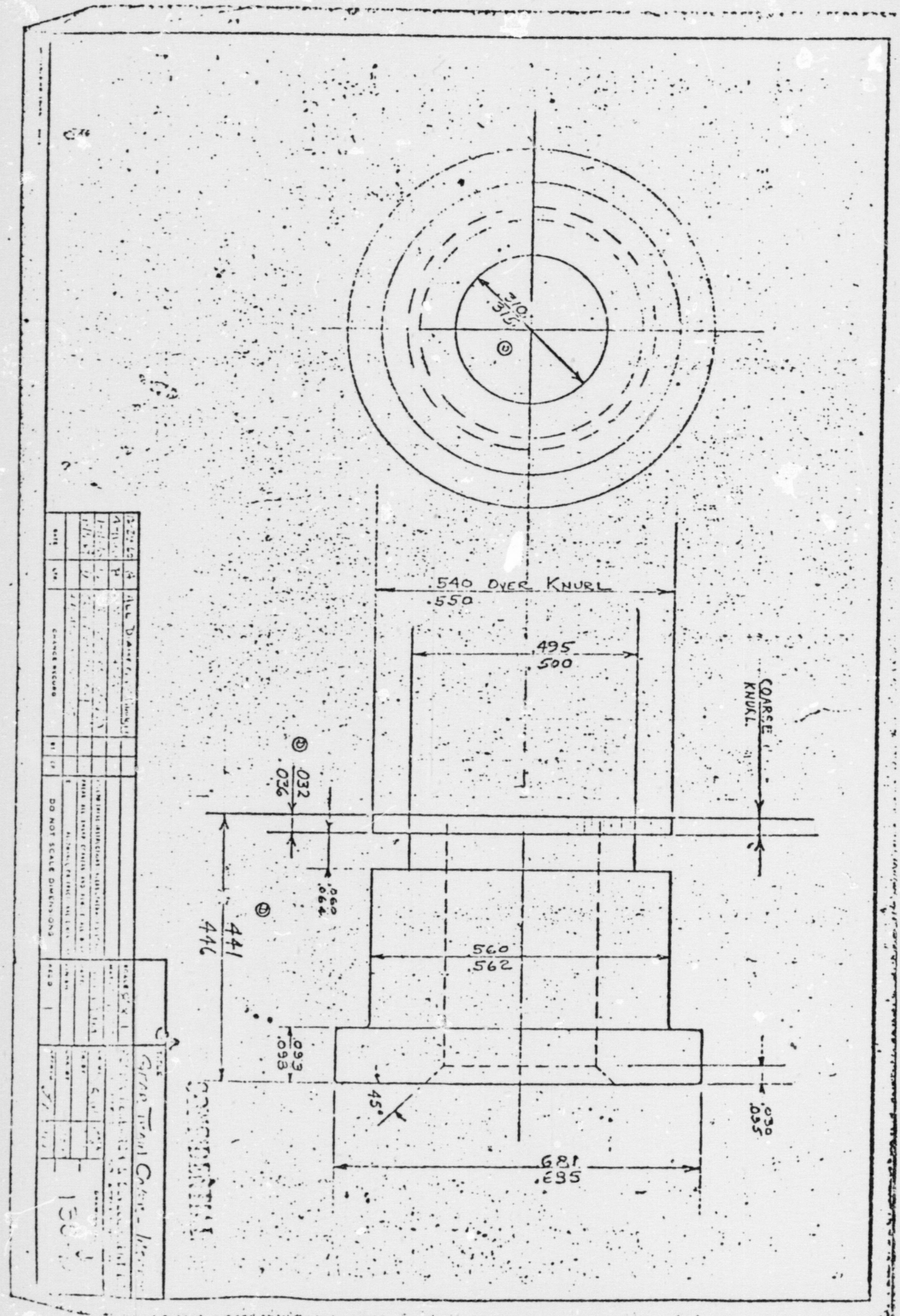


CONFIDENTIAL

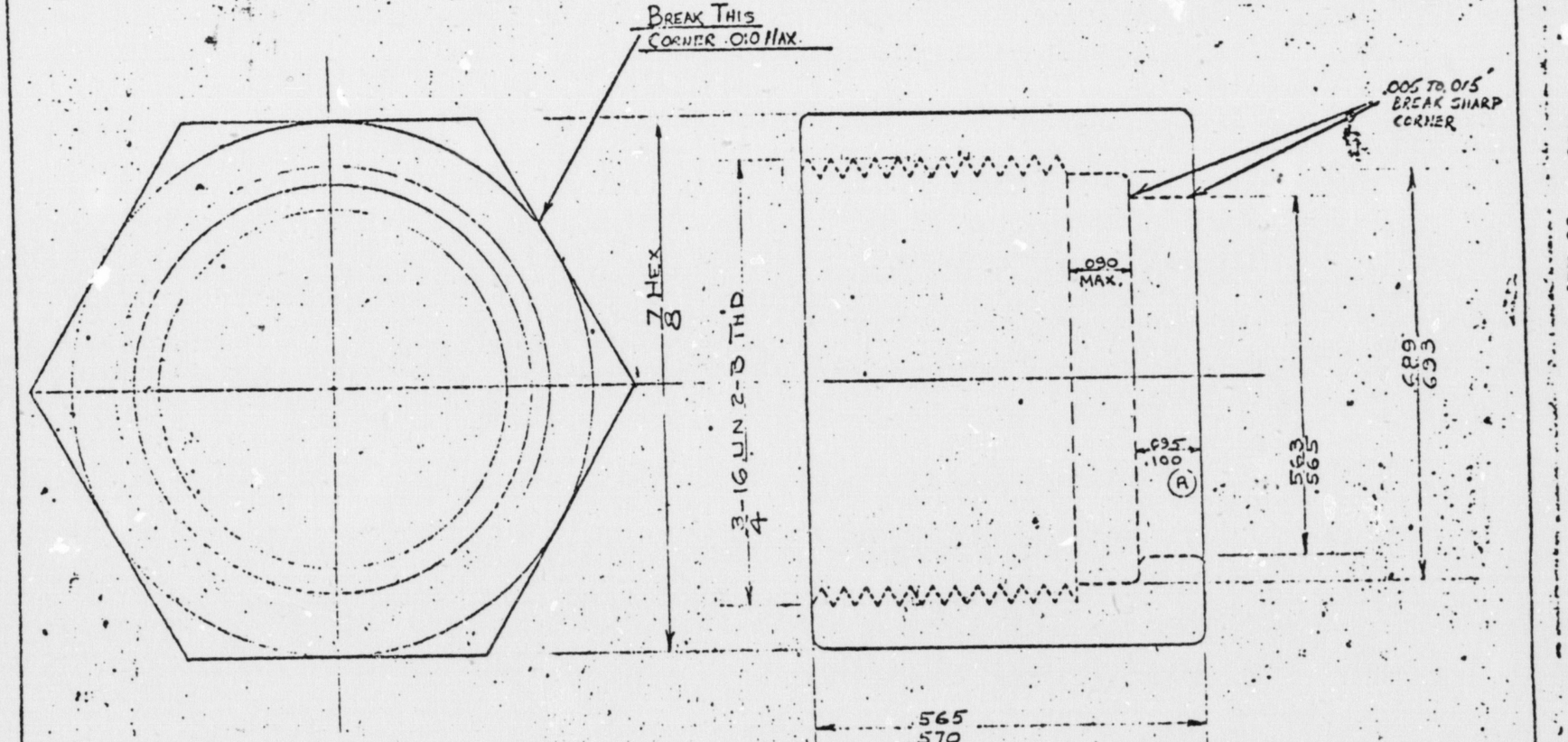
						SCALE 2 X 1	TITLE Gov. Housing Gasket
						MATL	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	A-2205	
11/3/53	B	Tolerance Revised			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	
10/7/53	A	Thickness Changed			± .010 ALLOWANCE ON MACHINED SURFACES	FINISH	
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	HEAT TREAT	

DRAW NO 129-B

E-22


.310
.315
.540 OVER KNURL
.550
.495
.500
COARSE KNURL
.032
.036
.060
.064
.441
.446
.560
.562
.093
.093
45°
.030
.035
.681
.685
DO NOT SCALE DIMENSIONS

[PX 23, p 62]



CONFIDENTIAL

NOTE:

NICKEL PLATE .0003

MIN. DIMENSIONS APPLY AFTER PLATING

DATE	LTR.	CHANGE RECORD	BY	CK
4-11-61	A	WAS 1½		

DO NOT SCALE DRAWING

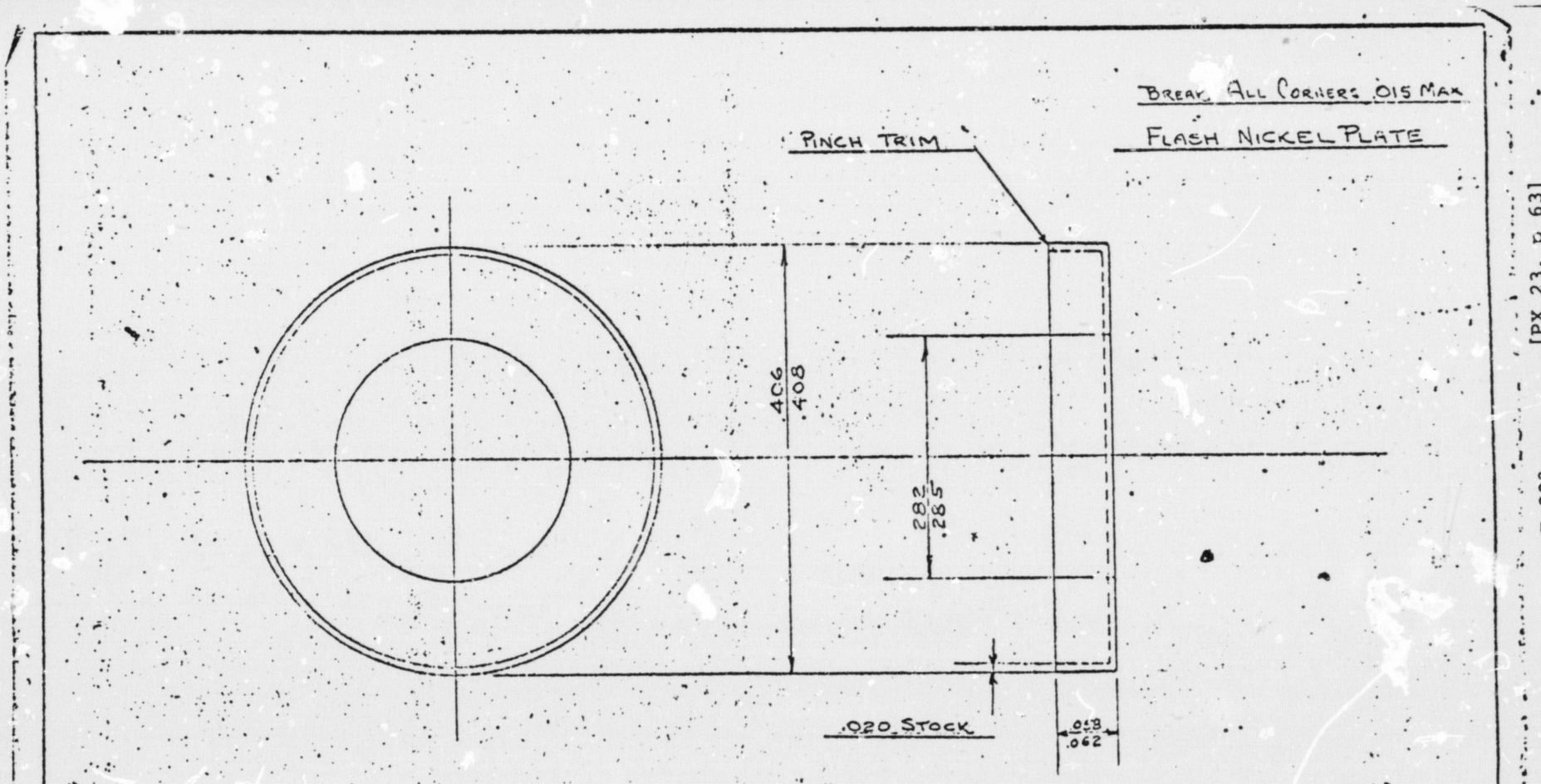
TITLE: HOUSING INSERT CLAMP #1

131-D

[PX 23, p 62]

E-229

[PX 23, p 63]

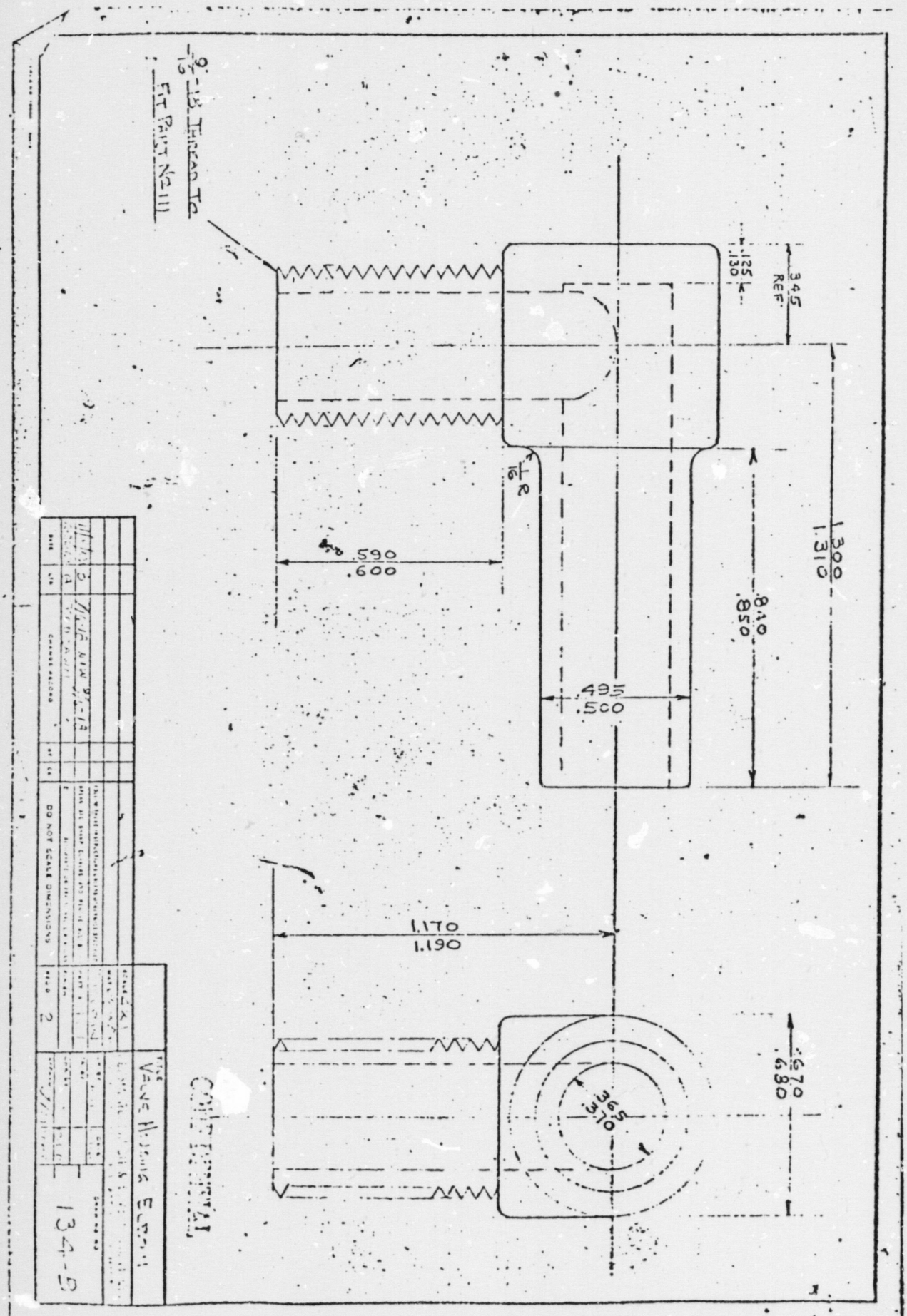


CONFIDENTIAL

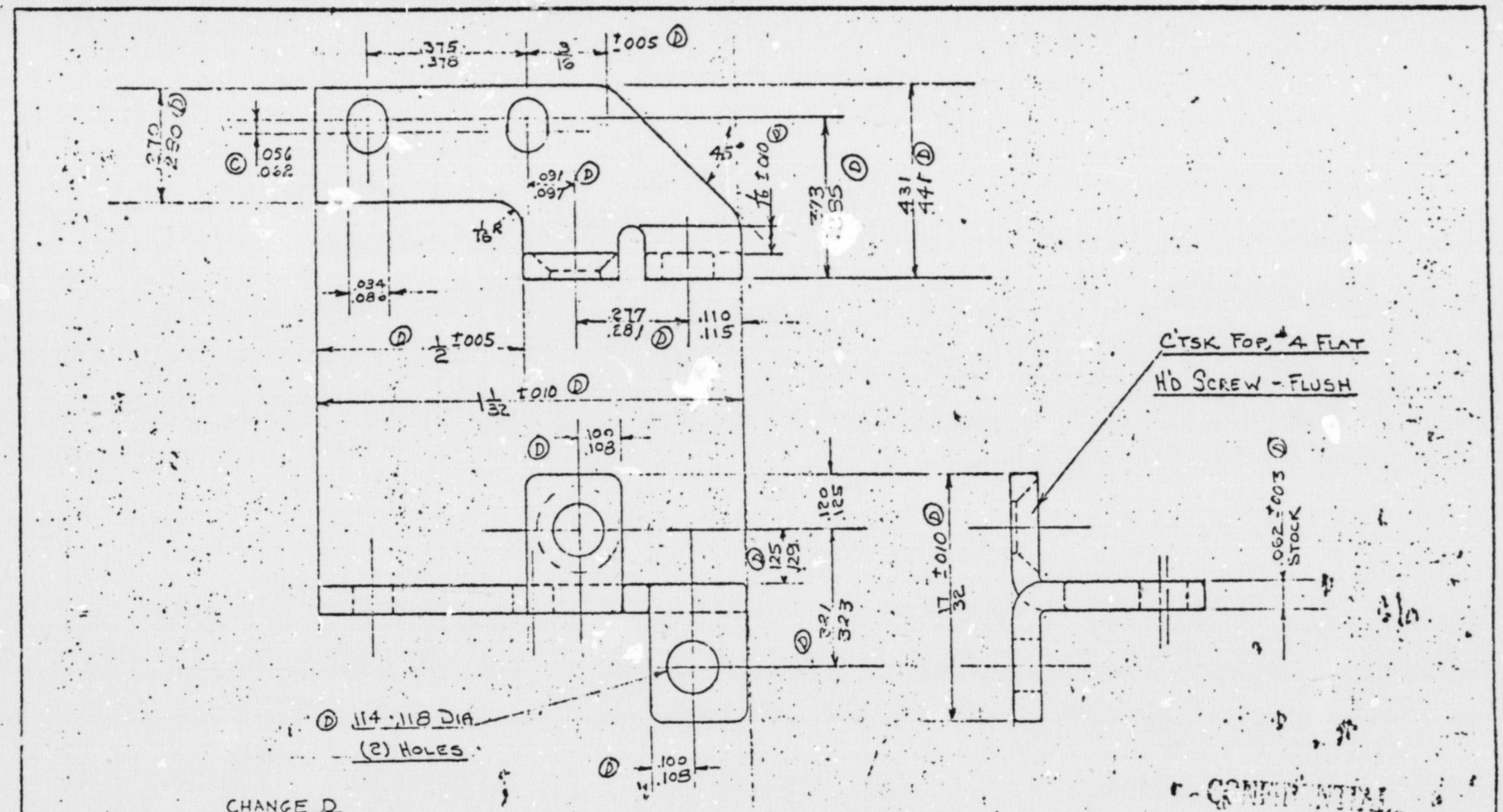
					SCALE (CM.) 1	TITLE VALVE THRUST FELT CUP
					MATL. PLATE	
				FOLLOW THESE INSTRUCTIONS UNLESS OTHWISE SPECIFIED		
1	8	STOCK SIZE CHANGED		BREAK ALL SHARP EDGES AND REMOVE ALL BURRS	PART	
2	A	REVISED FROM VELLUM TRACING		ALLOWANCE ON FRACTIONAL DIMENSIONS	PINION	132-P
DATE	LTR.	CHANGE RECORD	BY	DO NOT SCALE DIMENSIONS	REQD	

[PX 23, p 63]

E-230


9/16-18 THREAD TO
FIT PART NO 111
.345
REF.
.125
.130
1/2 R
.590
.600
.300
1.310
.840
.850
.495
.500
1.170
1.190
.670
.680
.510
.490
DO NOT SCALE DIMENSIONS
VALVE HOUSING ELBOW
134-5

[PX 23, p 65]



CHANGE D

ADD 1/2 ±.010 DIM
ADD ±.005 TO 1/2 & 7/16
.277-.281 WAS .272-.280
.431-.441 WAS .431-.435
.373-.355 WAS .373-.372
.272-.280 WAS .272-.274
.100-.108 WAS .100-.105 (2) PLACES

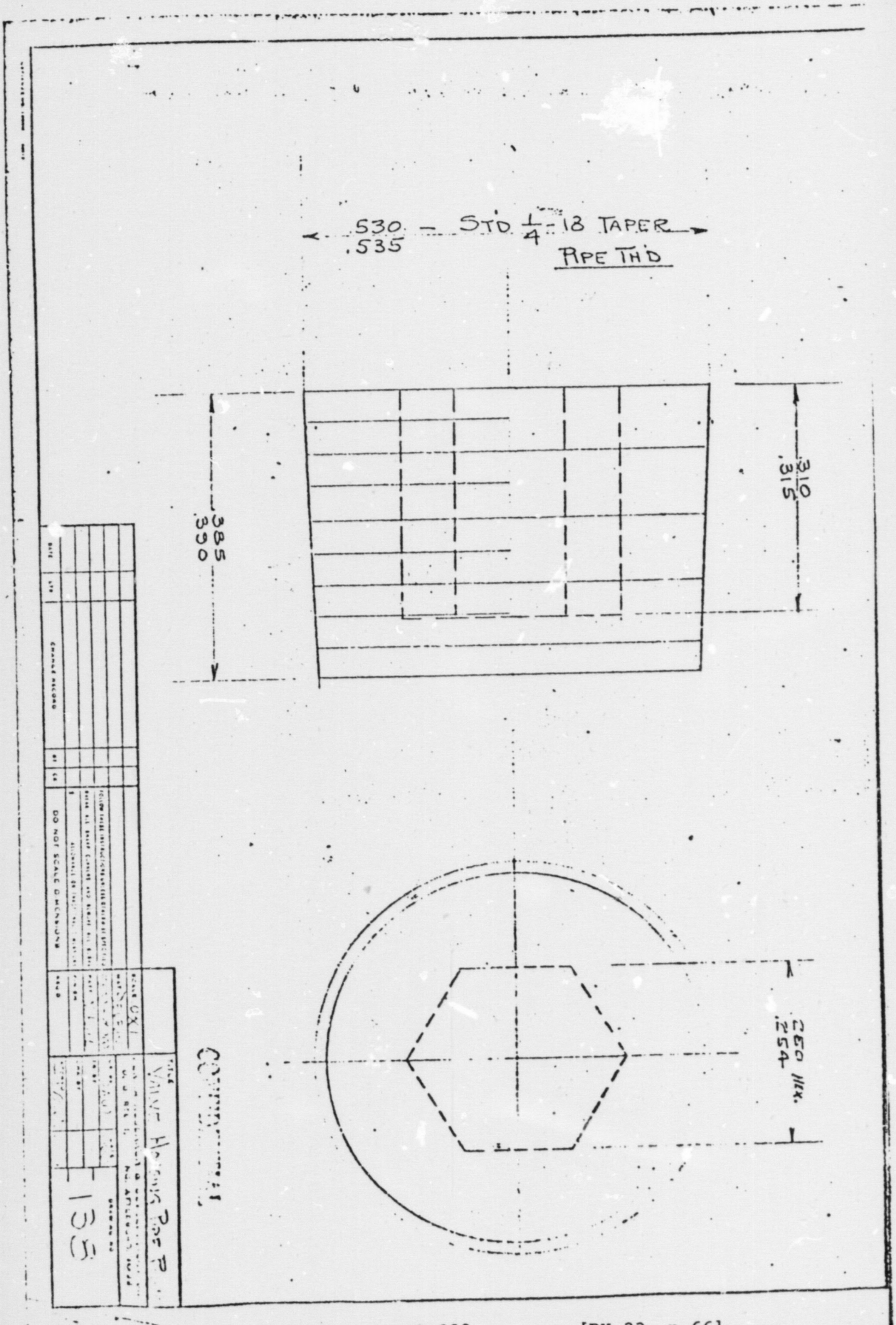
ADD ±.003 TO .062 STOCK
ADD ±.010 TO 1 1/3 & 17/32
.034-.080 WAS .093-.095
.125-.129 WAS .126-.129
.321-.323 WAS .320-.322
.114-.118 DIA WAS .115-.117

	D	SEE NOTE		FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	
	C			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	DATE
	B	Revised To Print Dtd		±.010 UNLESS OTHERWISE SPECIFIED	FINISH
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS

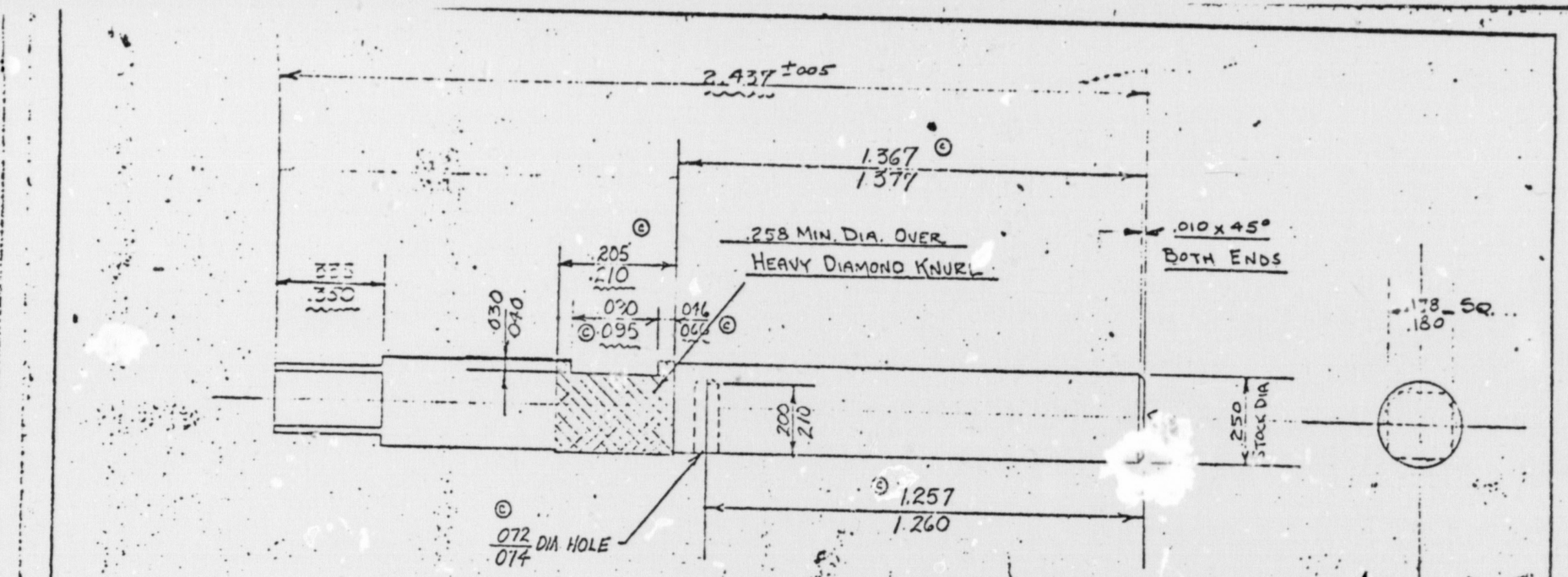
TITLE	Switch Bracket
SCALE	5X1
MATERIAL	
	ERMA RESEARCH & DEVELOPMENT CO.
DRAWING NO.	133-D

[PX 23, p 65]

E-232


.530
.535
STD 1/4-18 TAPER
PIPE TH'D
.385
.390
.310
.315
.250 MIN.
.254

[PX 23, p 67]



NOTE:
NICKEL PLATE (D)
.0003 MAXIMUM

CONFIDENTIAL

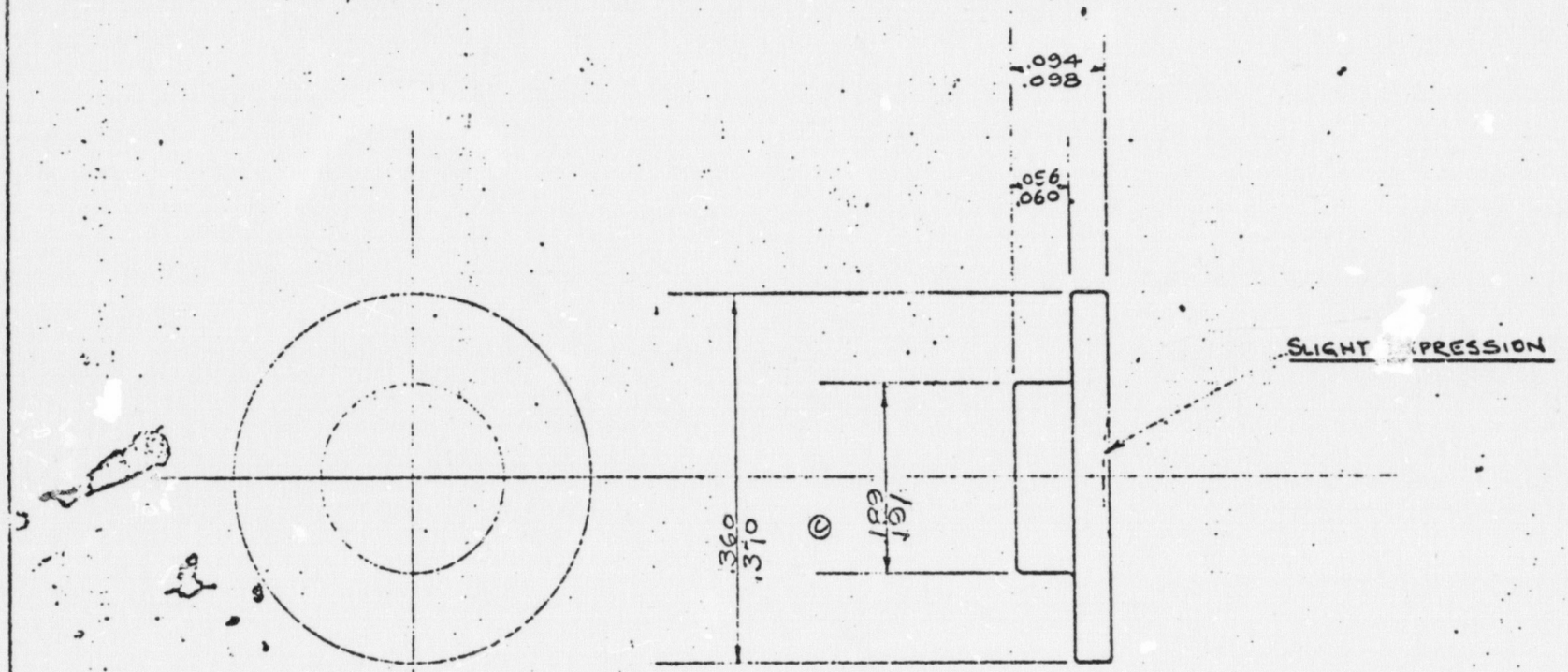
2-11-61	D	CADMIUM TO NICKEL			SCALE 4 x 1	TITLE: Anti-Skid Drive Gear Shaft
1-19-61	C				MATL	
2-6-60	B				FINISH	
4-28-59	A					
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	APPD

DRAWING NO. 137-D

[PX 23, p 67]

E-234

[PX 23, p 68]

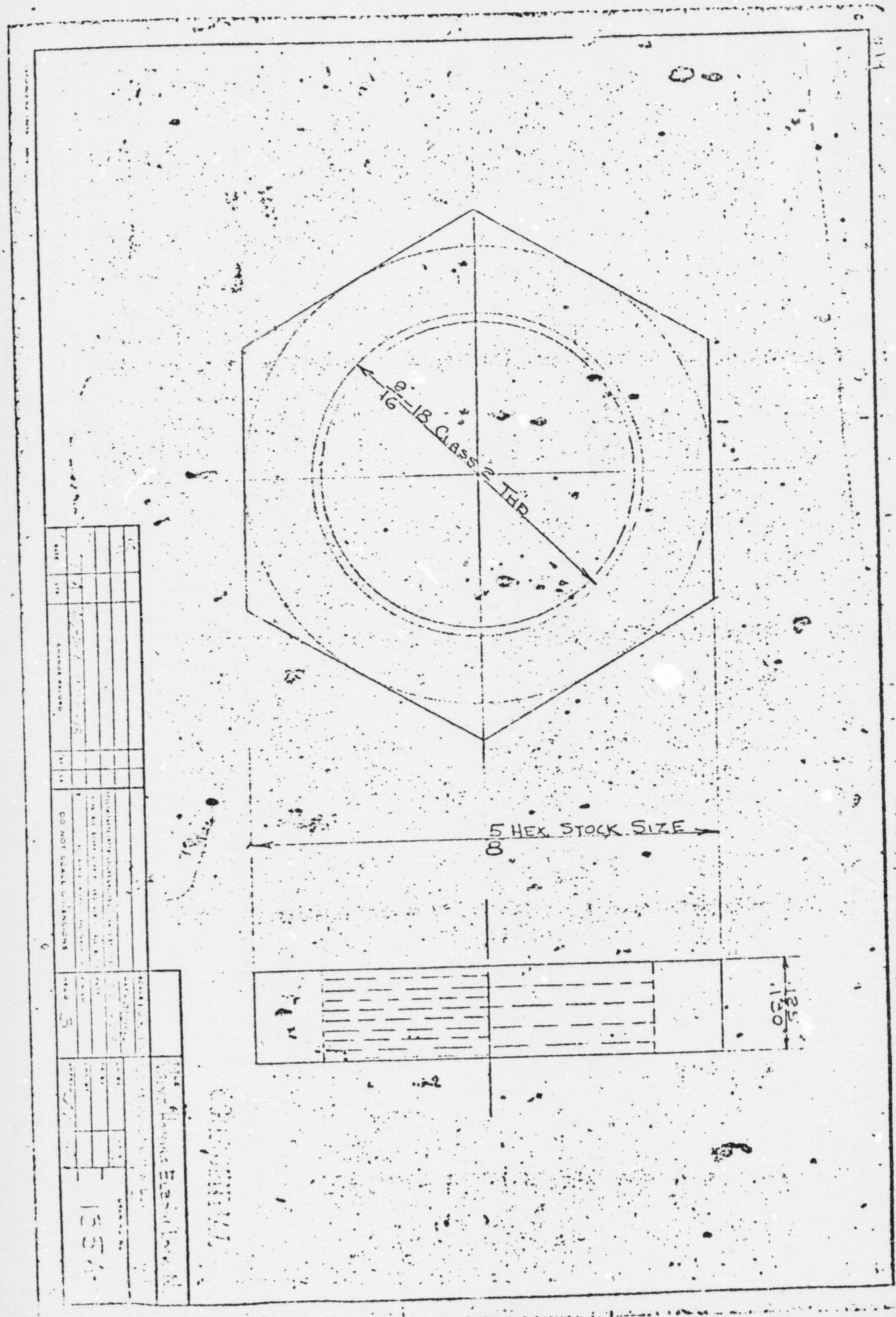


NOTE: DELRIN MUST BE OIL ANNEALED

						SCALE 10/1	TITLE CAM GEAR BUTTON
						MATL DELRIN	PERMA RESEARCH & DEVELOPMENT CO.
10/8/92	C	109/101 WAS 107/192			FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	FINISH	DRAWN BY [illegible] DATE 10/8/92
2/10/92	B	ADD OIL ANNEAL			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PART NO.	CHK'D BY DATE
1/2/92	A	[illegible]			DIMENSIONS OR SPECIFICATIONS APPLY	FINISH	APPD BY DATE
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D	DRAWING NO 136-C

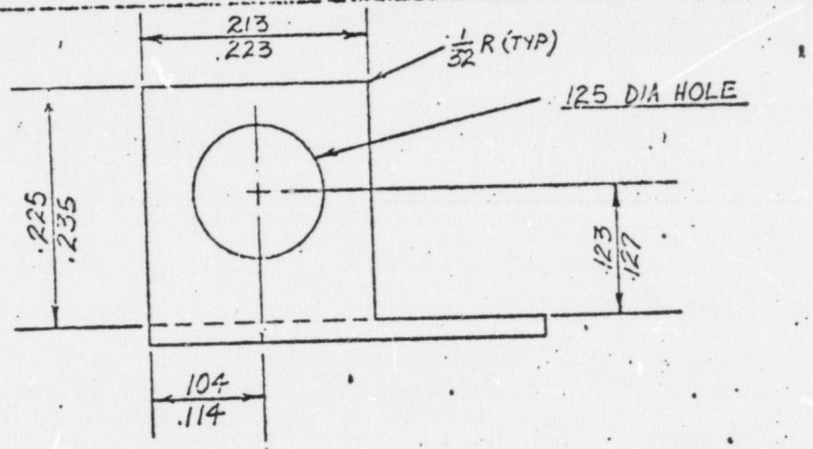
[PX 23, p 68]

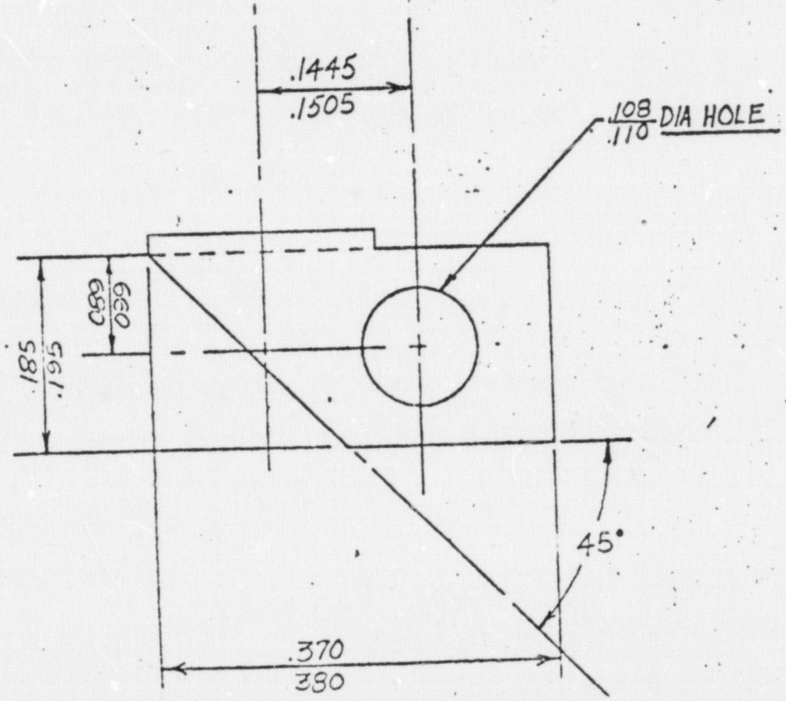
E-235



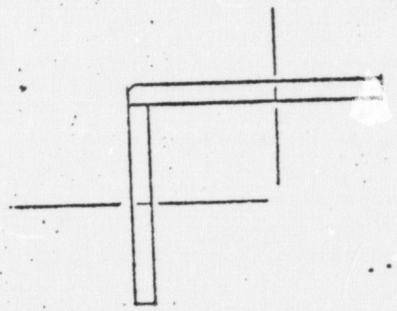
[PX 23, p 69]

[PX 23, p 70]





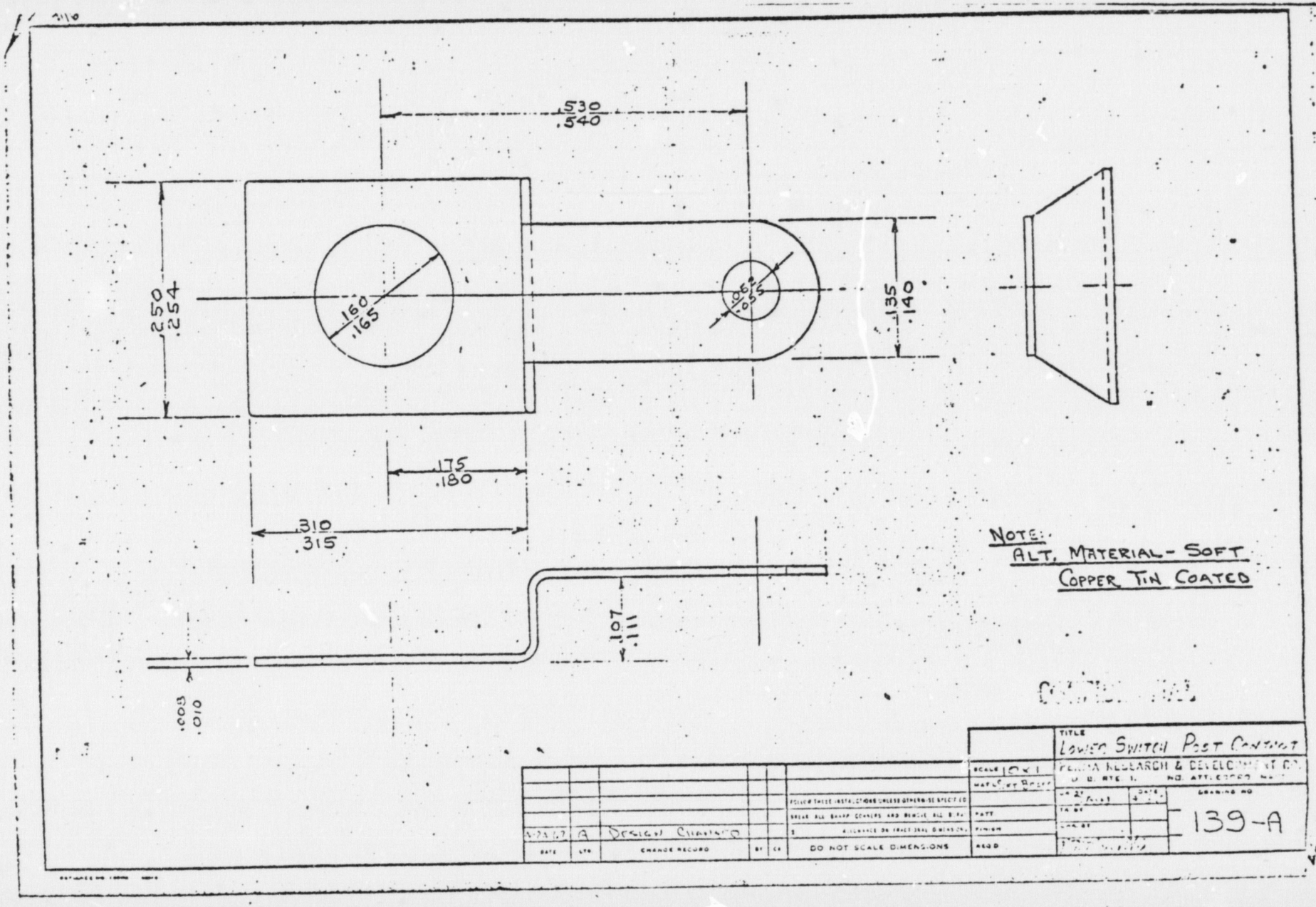
NOTE:
MATERIAL .018-.022 THICKNESS BRASS



CONFIDENTIAL

	TITLE UPPER SWITCH CONTACT	
SCALE 10X1	PERMA ELECTRONICS AND EQUIPMENT CO NEWTONVILLE MASS	
MAT BRASS	DR BY RDM	140-D
FATT	CH BY	
FIN	APPD	
QUAD 1		

[PX 23, p 71]



[PX 23, p 71]

E-238

[PX 23, p 72]

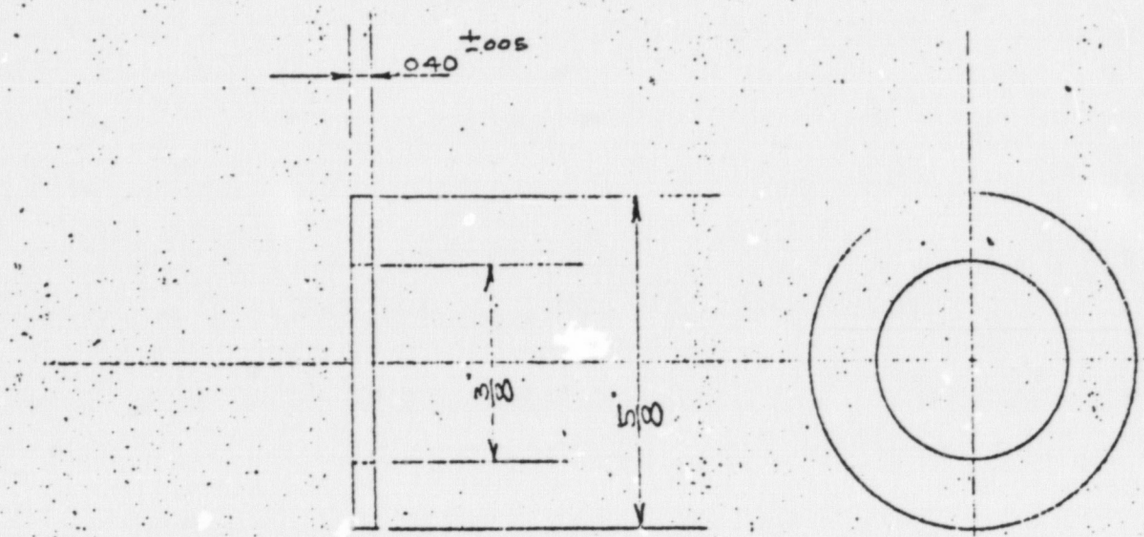
CONFIDENTIAL

						SCALE 10 X 1	TITLE LOWER FLAT CONTACT
						MAT'L SOFT PVC	PERMA RESEARCH & DEVELOPMENT CO
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DR BY A.L.J. DATE
					REMOVE ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	CH BY DATE
					ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH	APP BY DATE
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MATL	DRAWING NO 141

[PX 23, p 72]

E-239

[PX 23, p 73]



NOTE:
MAKE FROM
BARSEAL A-8209
OR EQUIVELENT

CONFIDENTIAL

DATE	LTR.	CHANGE RECORD	BY	CK.
11/2/43	A	BARSEAL OR EQUIVALENT		

DO NOT SCALE DIMENSIONS

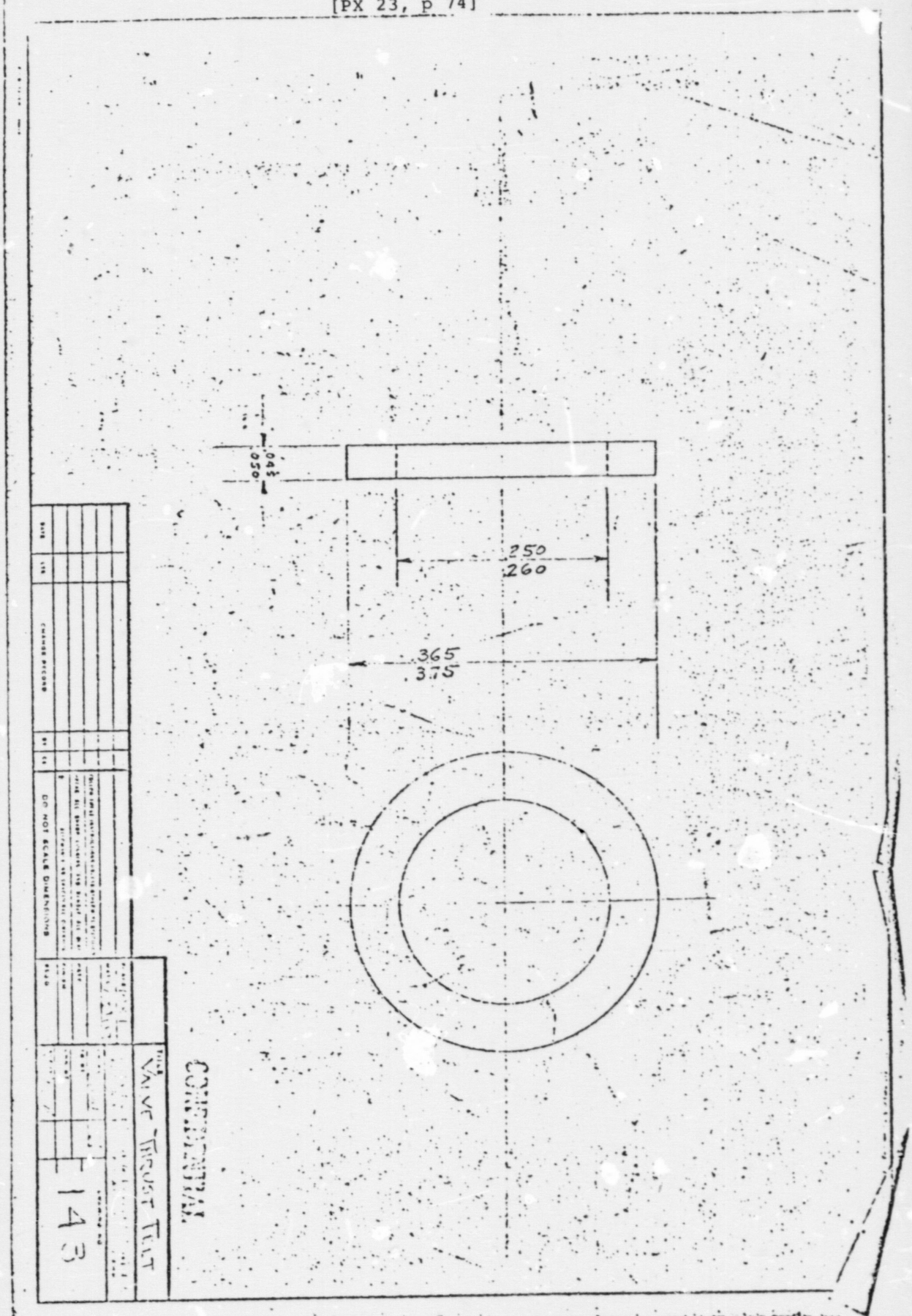
FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED
BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS
± ALLOWANCE ON FRACTIONAL DIMENSIONS

SCALE 4 x 1
MATERIAL SEE NOTE

TITLE VALVE HOLE PLUG GASKET

DRAWING NO 142-A

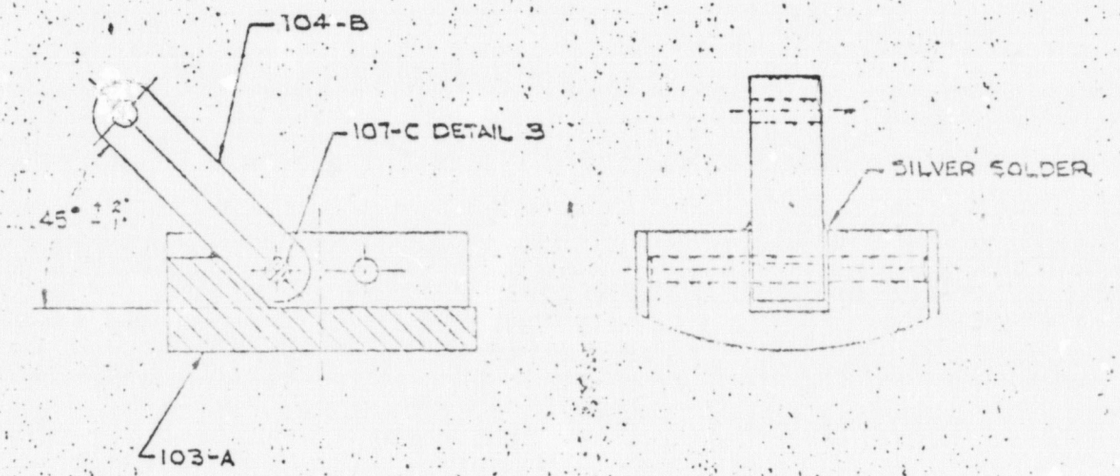
[PX 23, p 73]


.050
.043
250
260
365
375
DO NOT SCALE DIMENSIONS
VALVE THRUST TEST
143

[PX 23, p 75]

E-242

[PX 23, p 75]

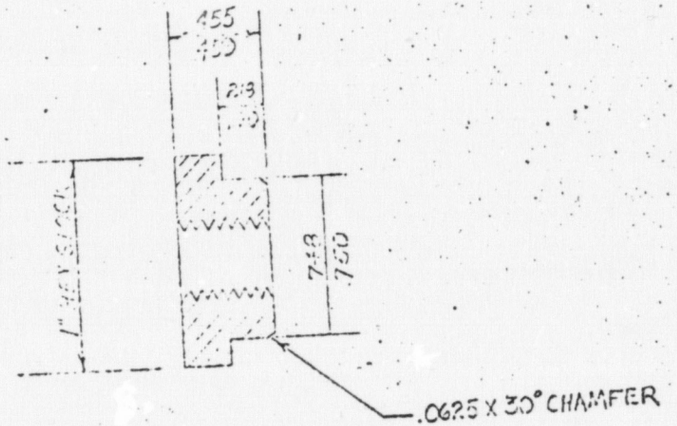


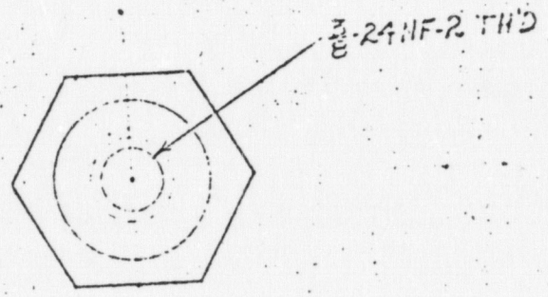
CONFIDENTIAL

| | | |

MATERIAL: COLD ROLL STEEL - MACH 5

BREAK ALL CORNERS .015 MAX





[PX 23, p 76]

E-243

[PX 23, p 76]

[PX 23, p 77]

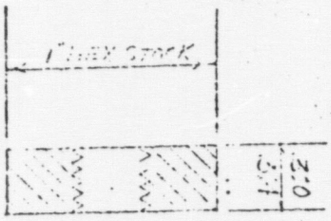
NOTE. MATERIAL- COLD ROLLED STEEL-ANCH 5

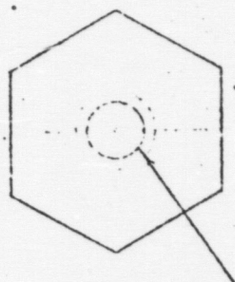
$\frac{7}{8}$-14 NF-2 THD

JAM NUT

E-244

[PX 23, p 77]





MATERIAL: COLD ROLL STEEL

BREAK ALL EDGES .010 MAX

[PX 23, p 79]

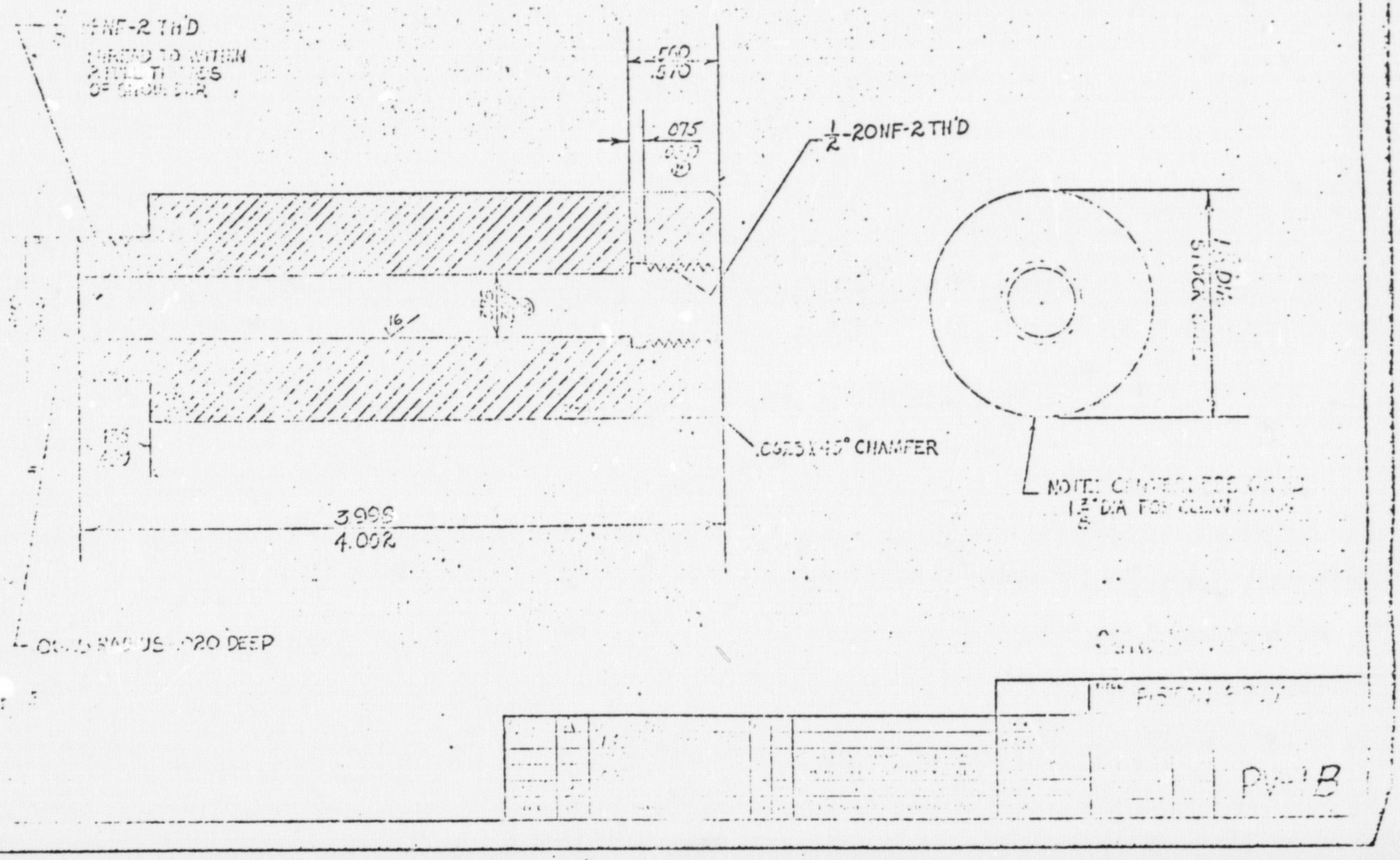
MATERIAL: COLD ROLL STEEL - INCH 5.

BREAK ALL CORNERS .015 MAX

PISTON STOP

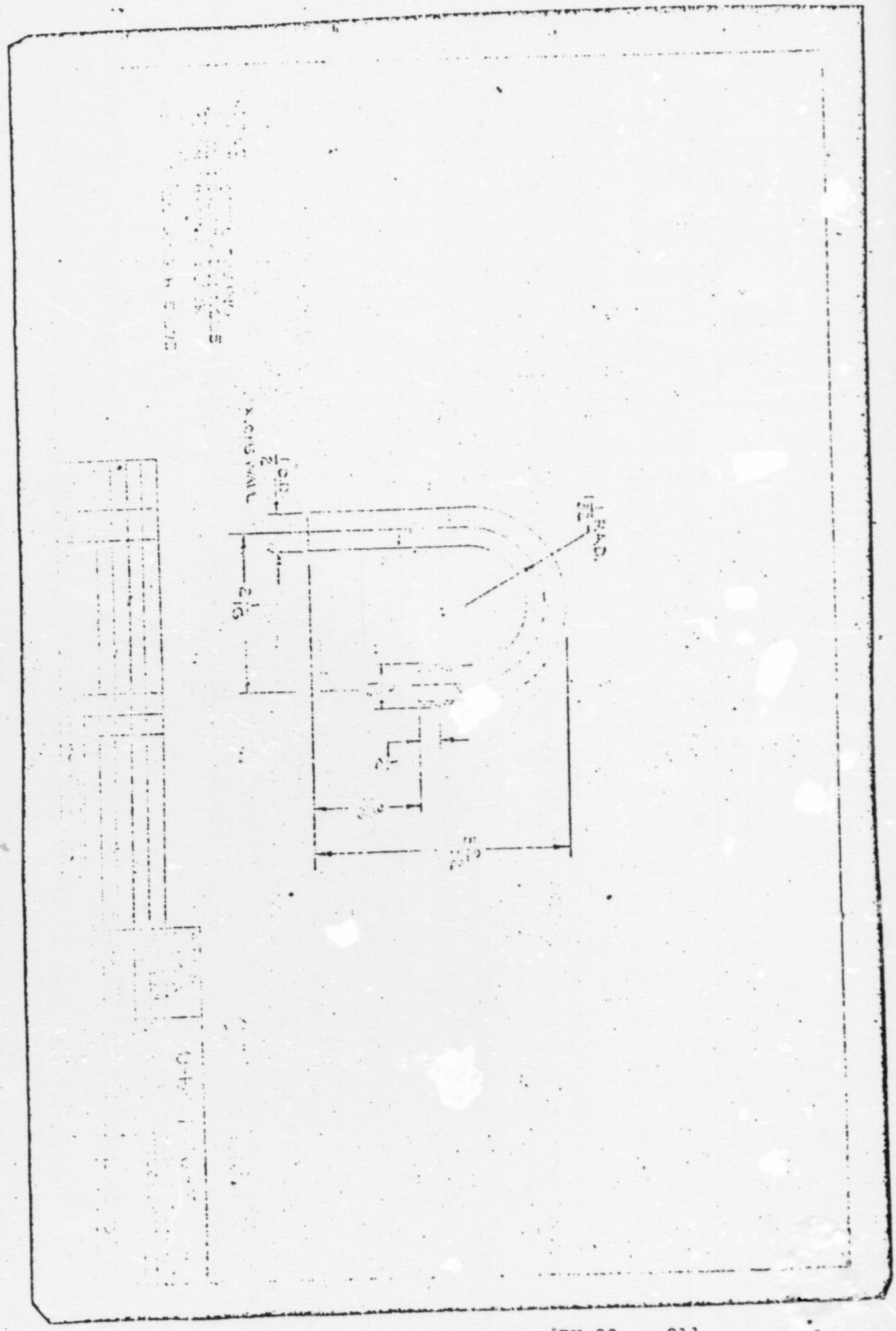
[PX 23, p 79]

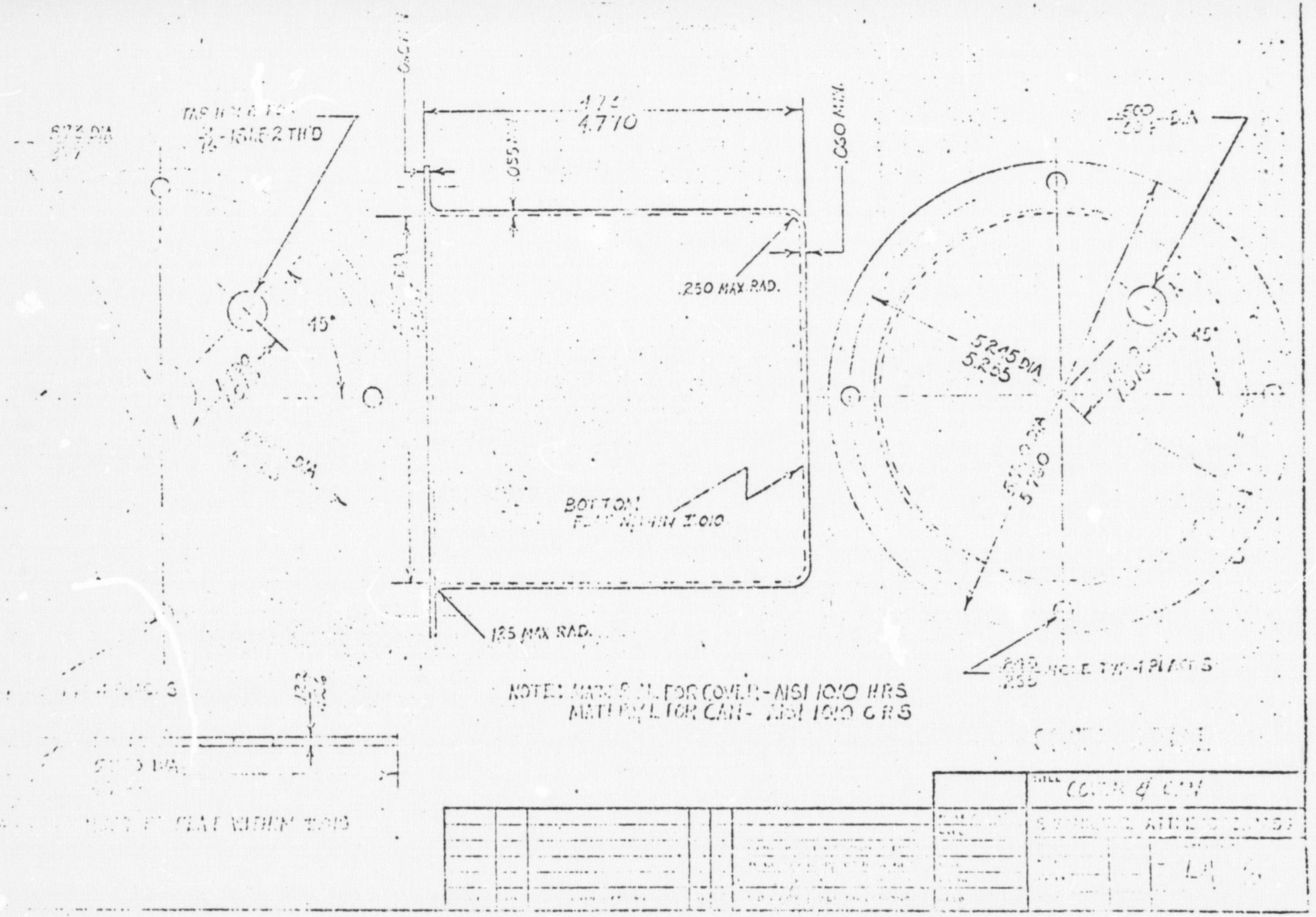
[PX 23, p 80]



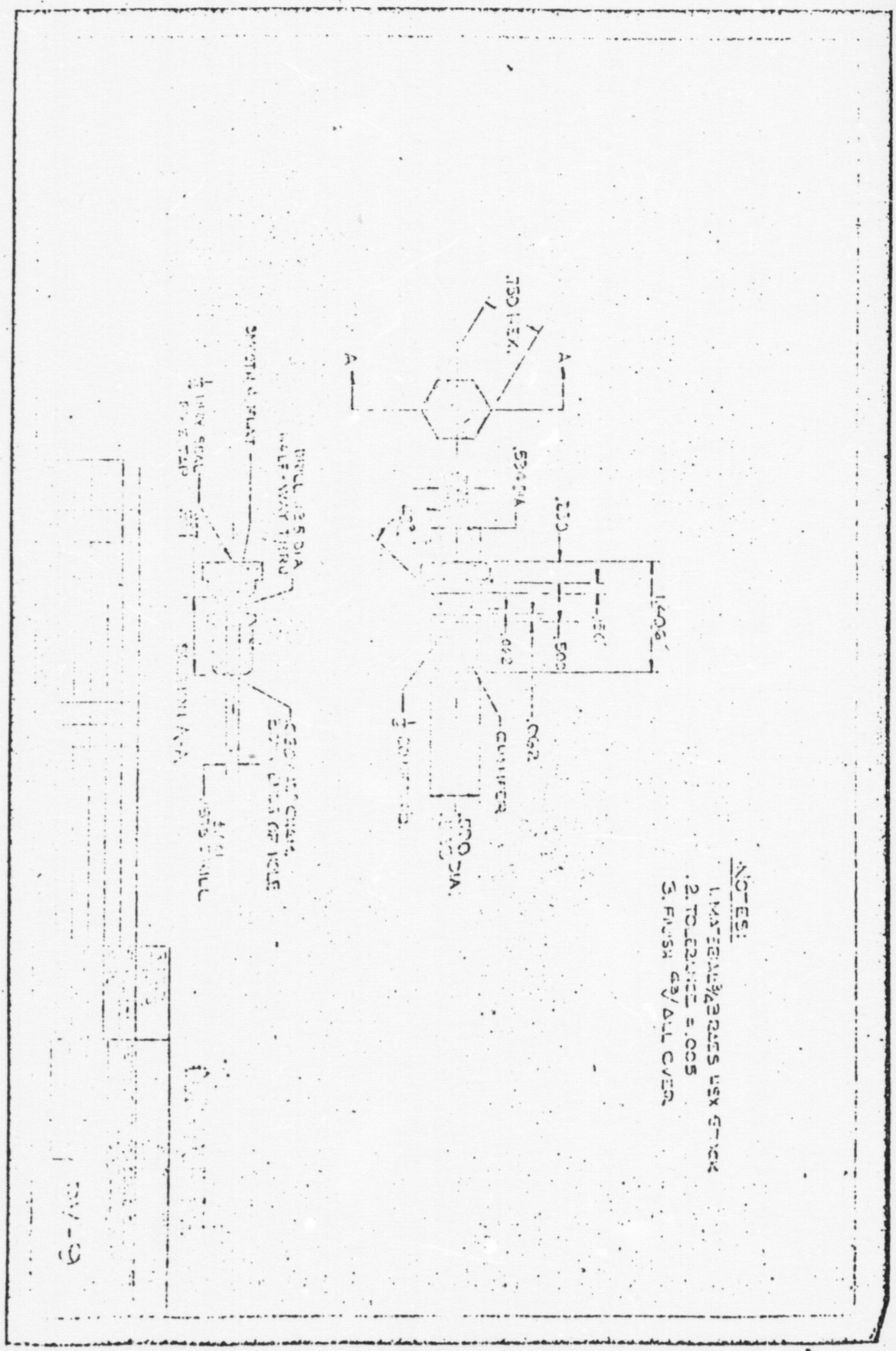
[PX 23, p 80]

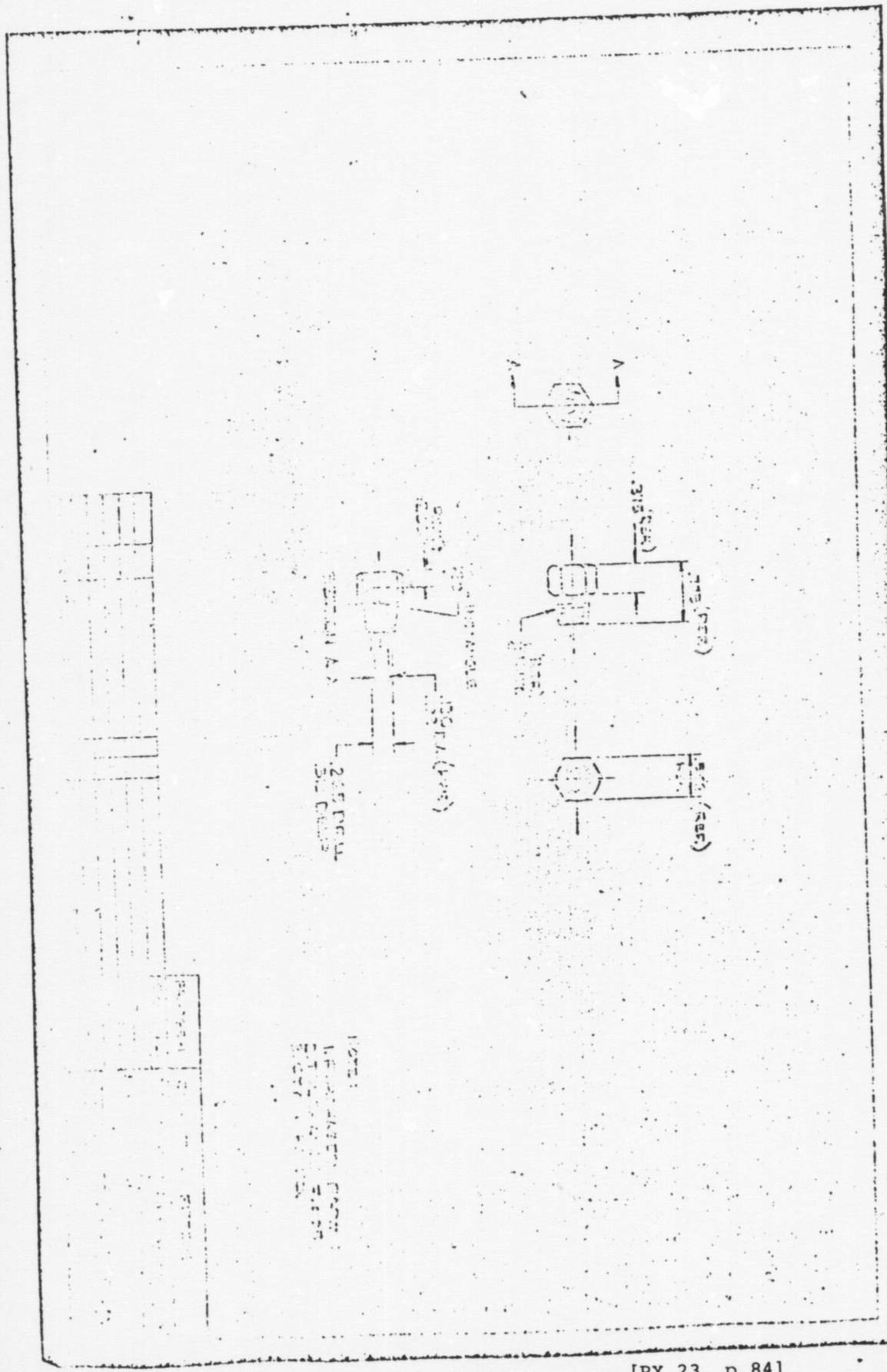
E-247

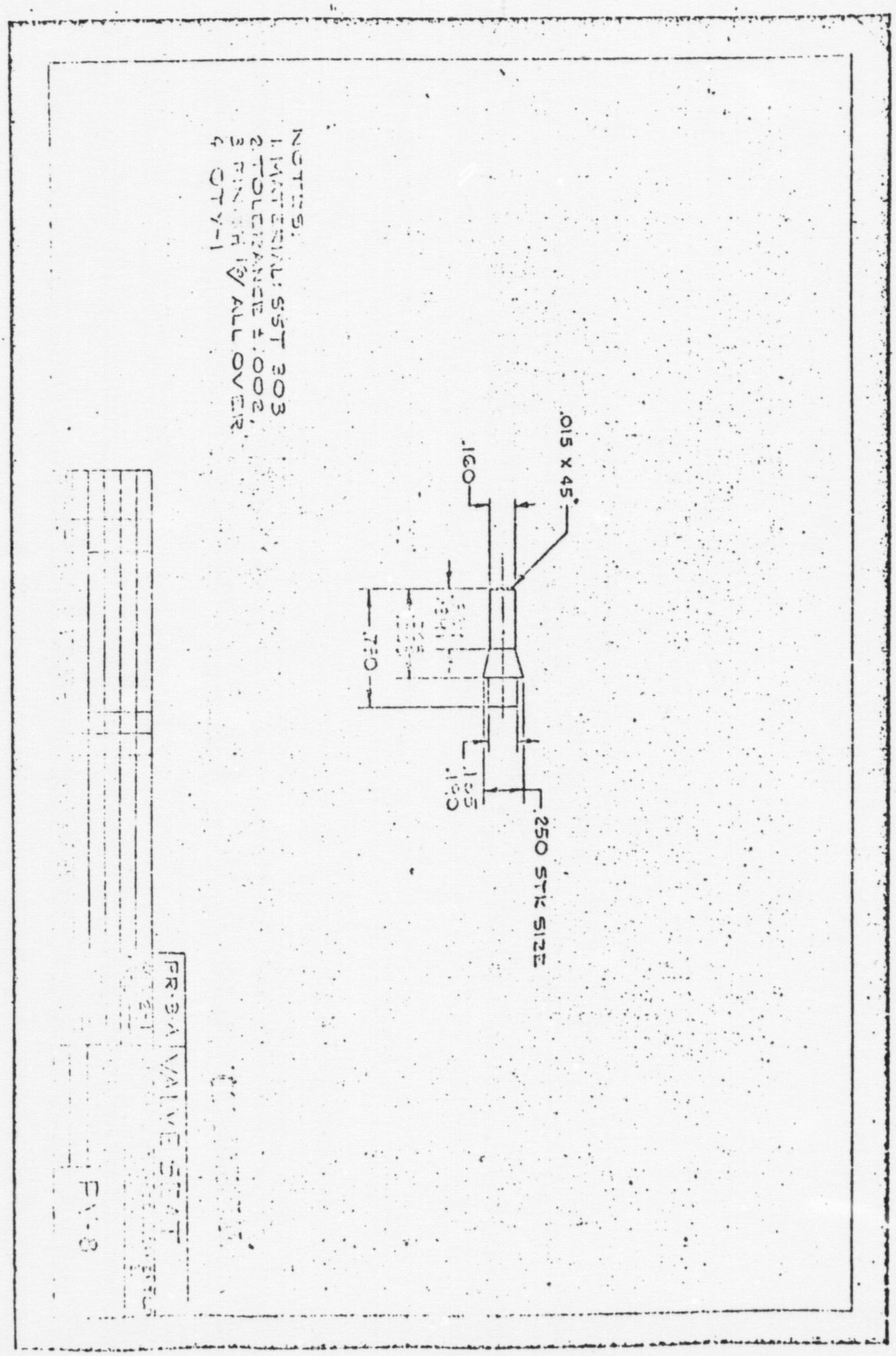


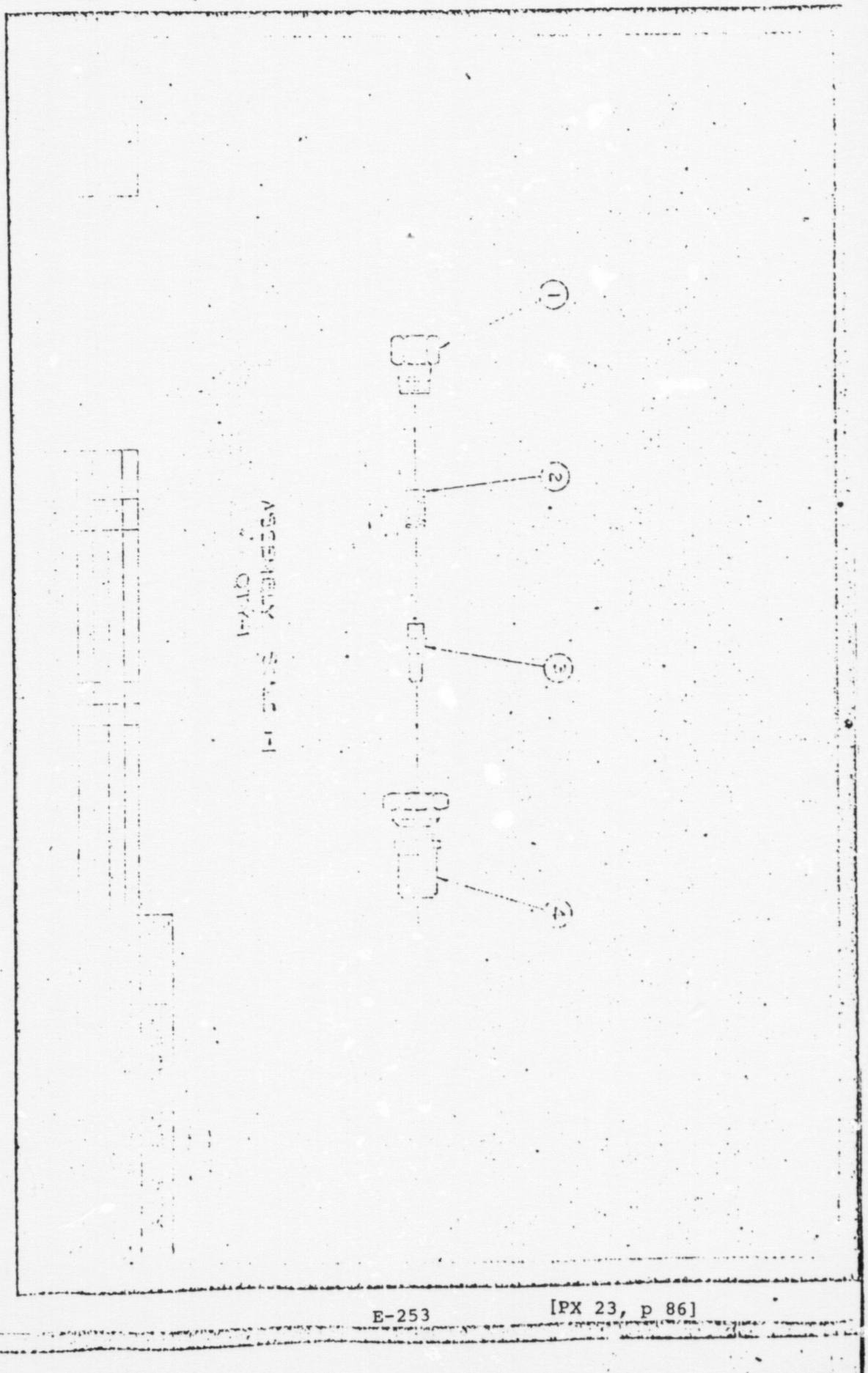

4.770
.250 MAX RAD.
.125 MAX RAD.
.050 MIN.
.055
15°
45°
5.245 DIA
5.255
NOTES: MATERIAL FOR COVER - AISI 1010 HRS
MATERIAL FOR CAN - AISI 1010 CRS

[PX 23, p 82]

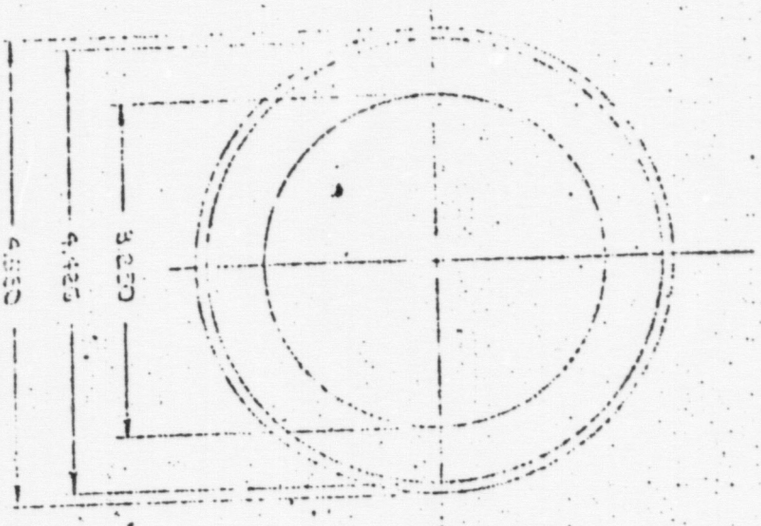


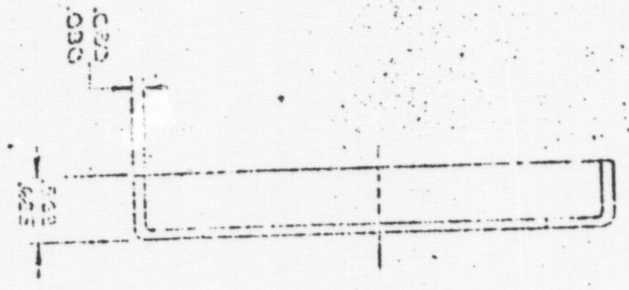



.015 x 45°
.160
.250 STK SIZE



[PX 23, p 86]


6.625
4.625
3.250

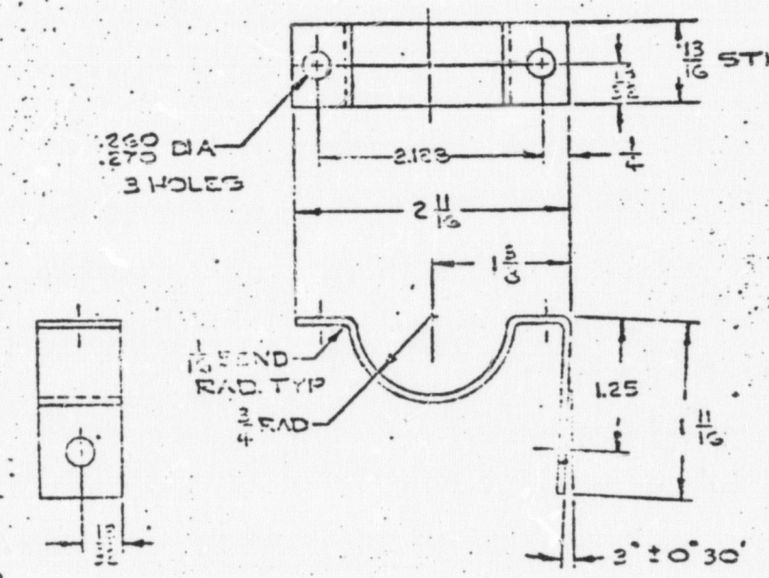

.250
.000
.188

[PX 23, p 88]

[PX 23, p 88]

E-255

[PX 23, p 89]

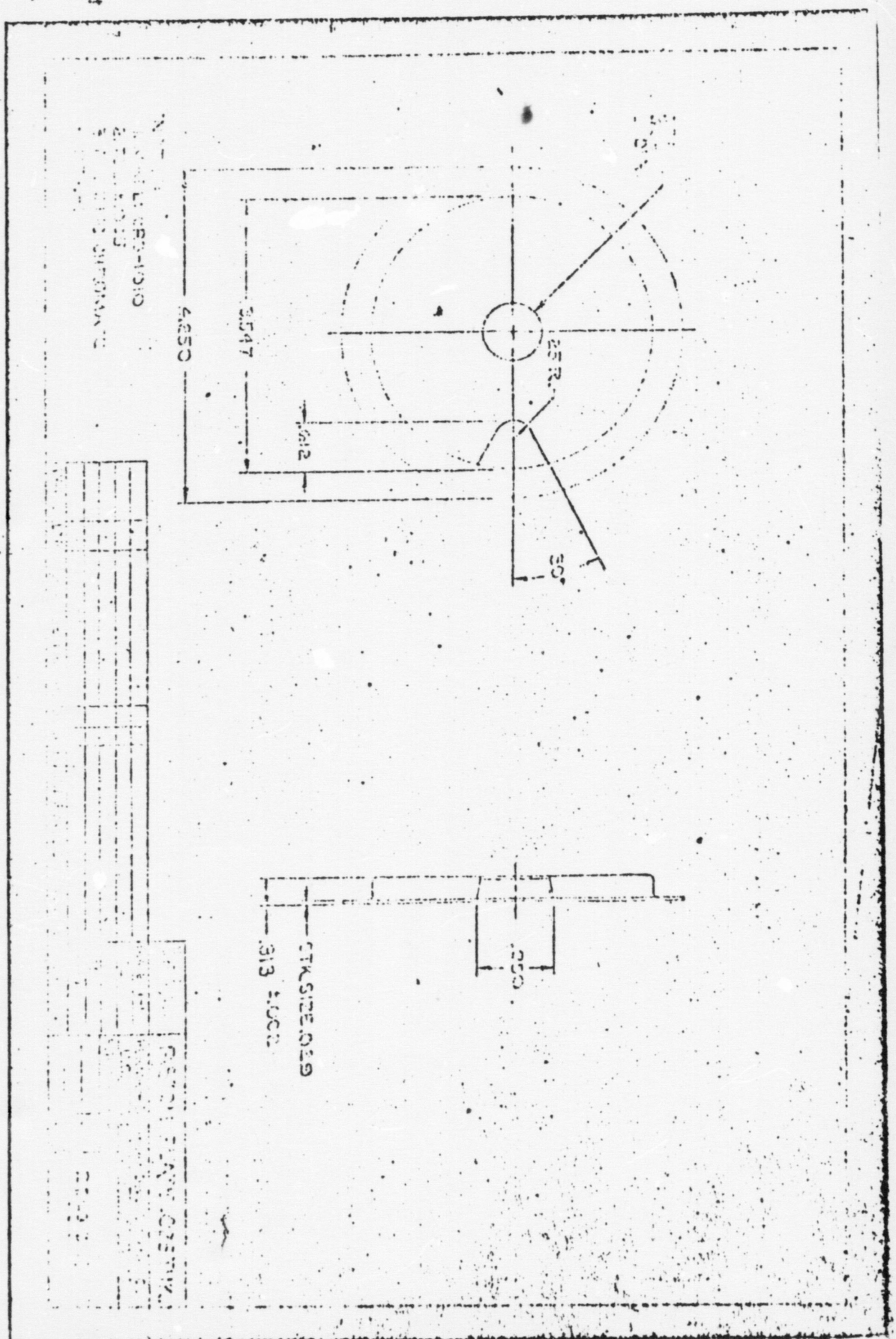


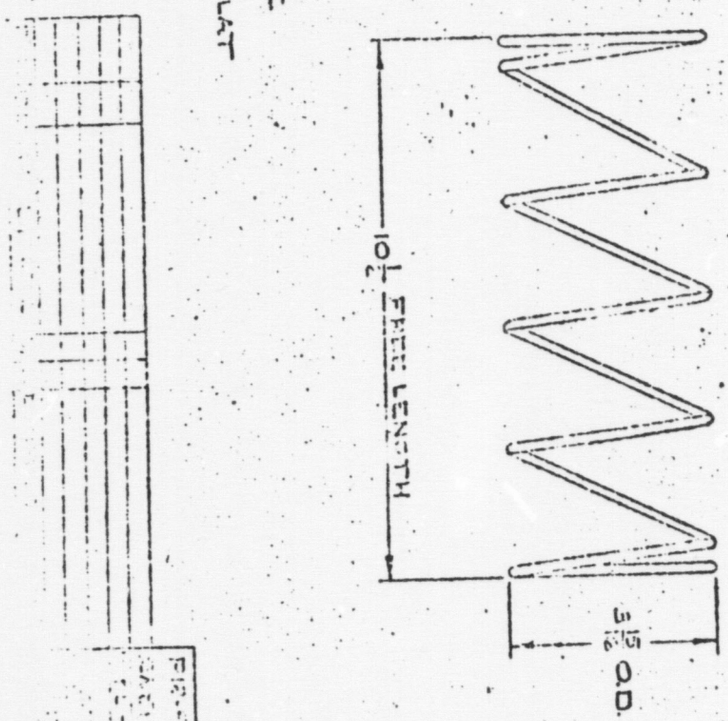
NOTES:
1. MATERIAL: .062 THK CRS-1010
2. TOLERANCES: FRAC ±1/64 DEC ±.005
3. FINISH: ZINC CHROMATE
4. REMOVE BURRS & SHARP EDGES
5. QTY-2

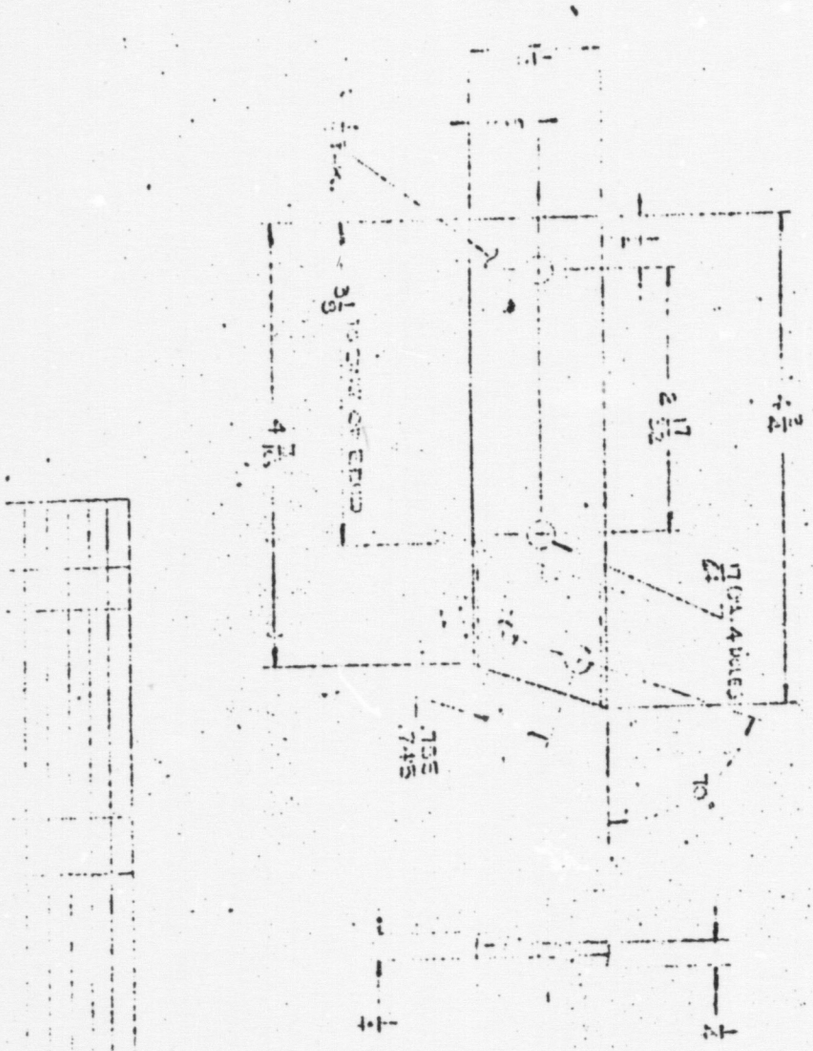
	PR-7A	ANTI SKID MTG. ET

[PX 23, p 89]

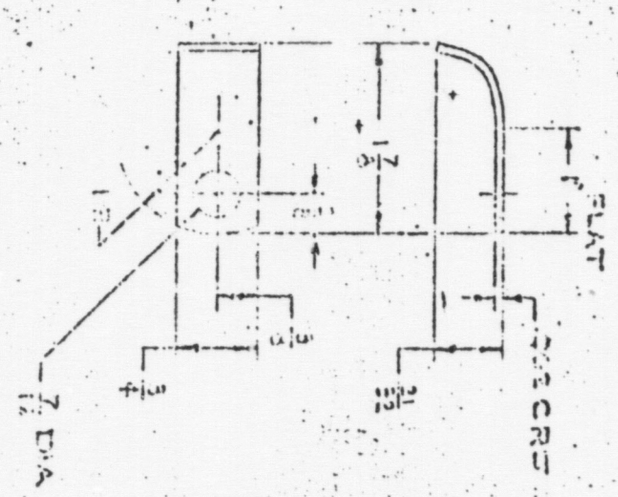
E-256

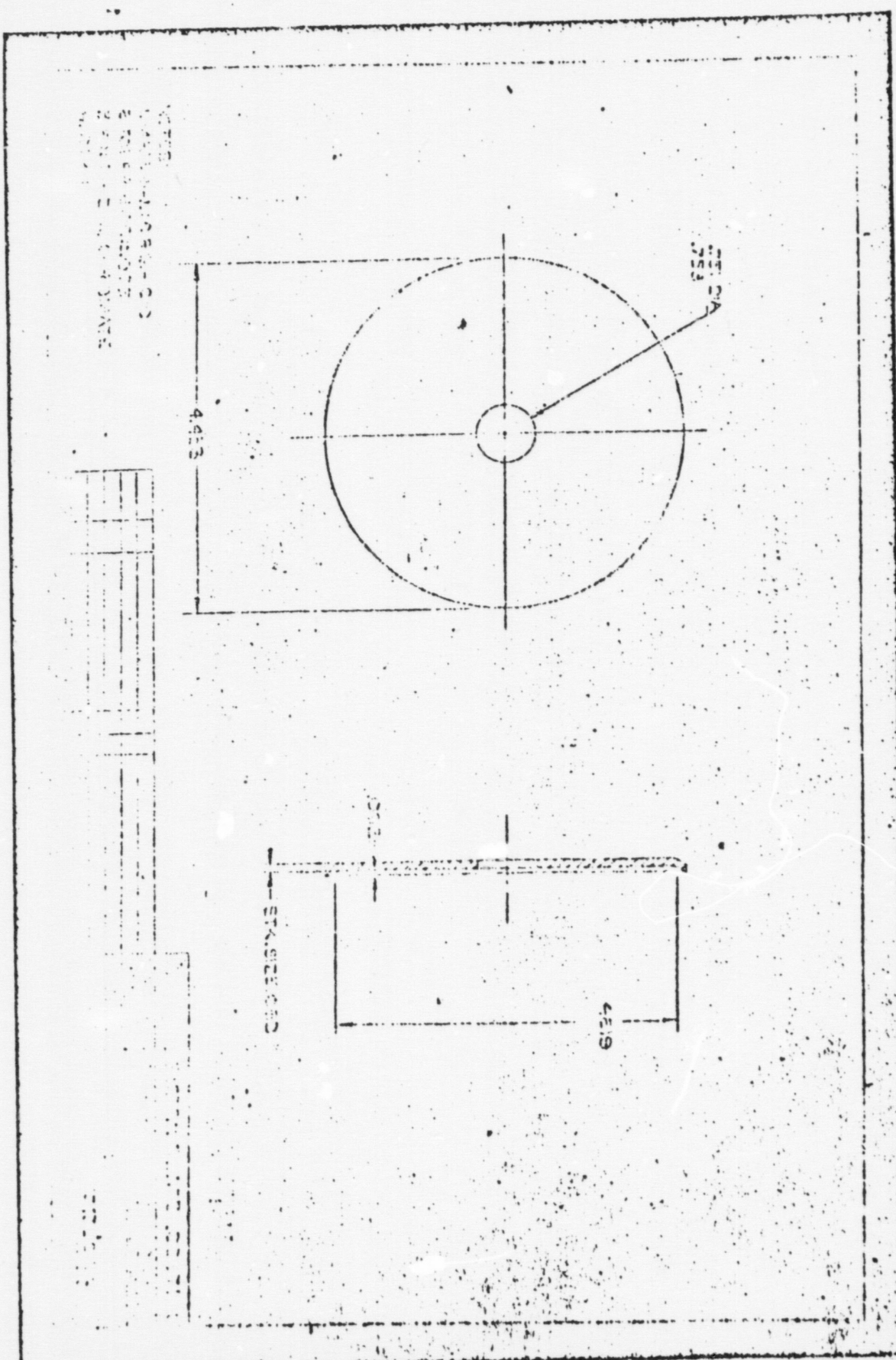


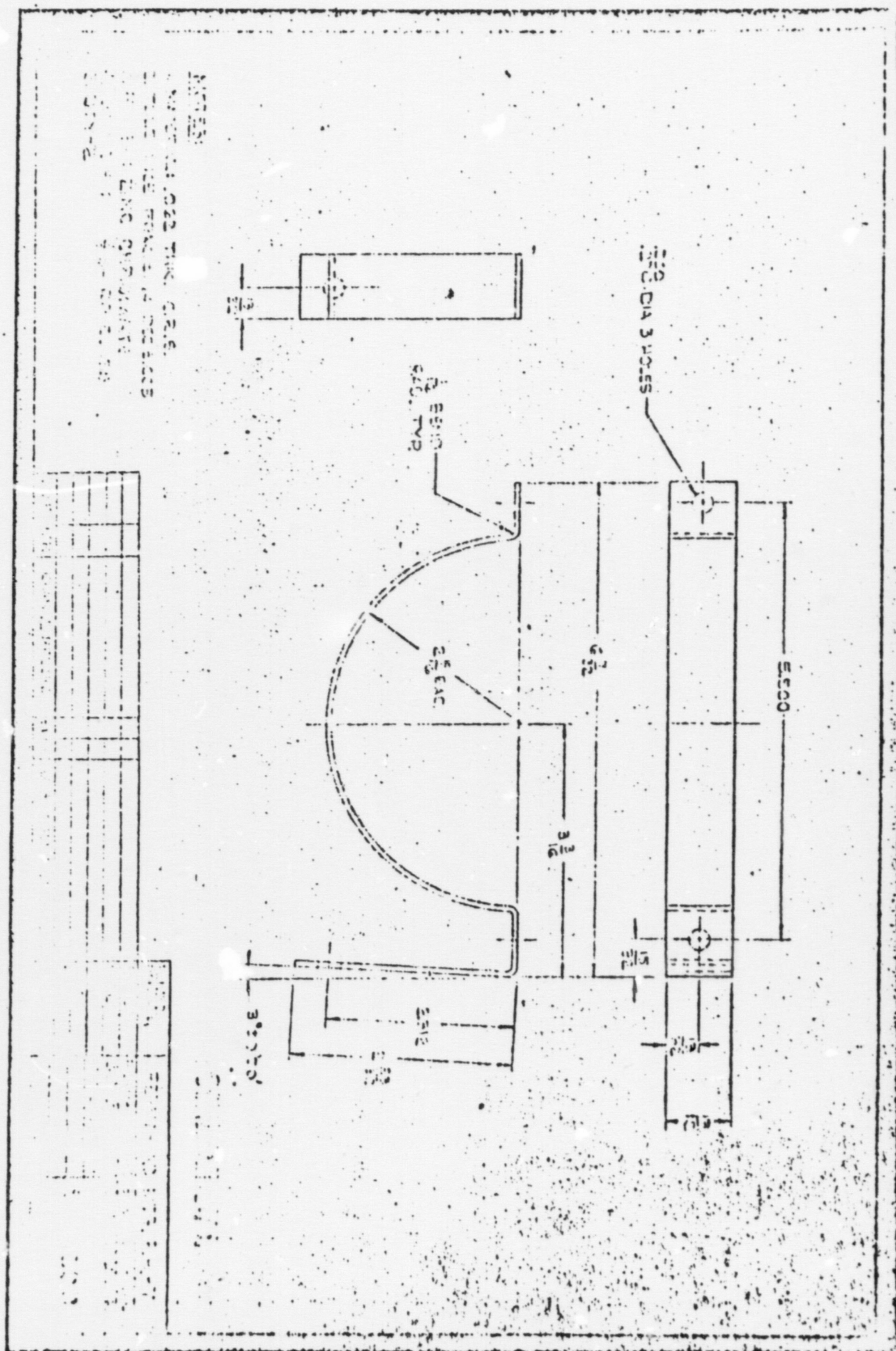

10½ FREE LENGTH
5/16 O.D.

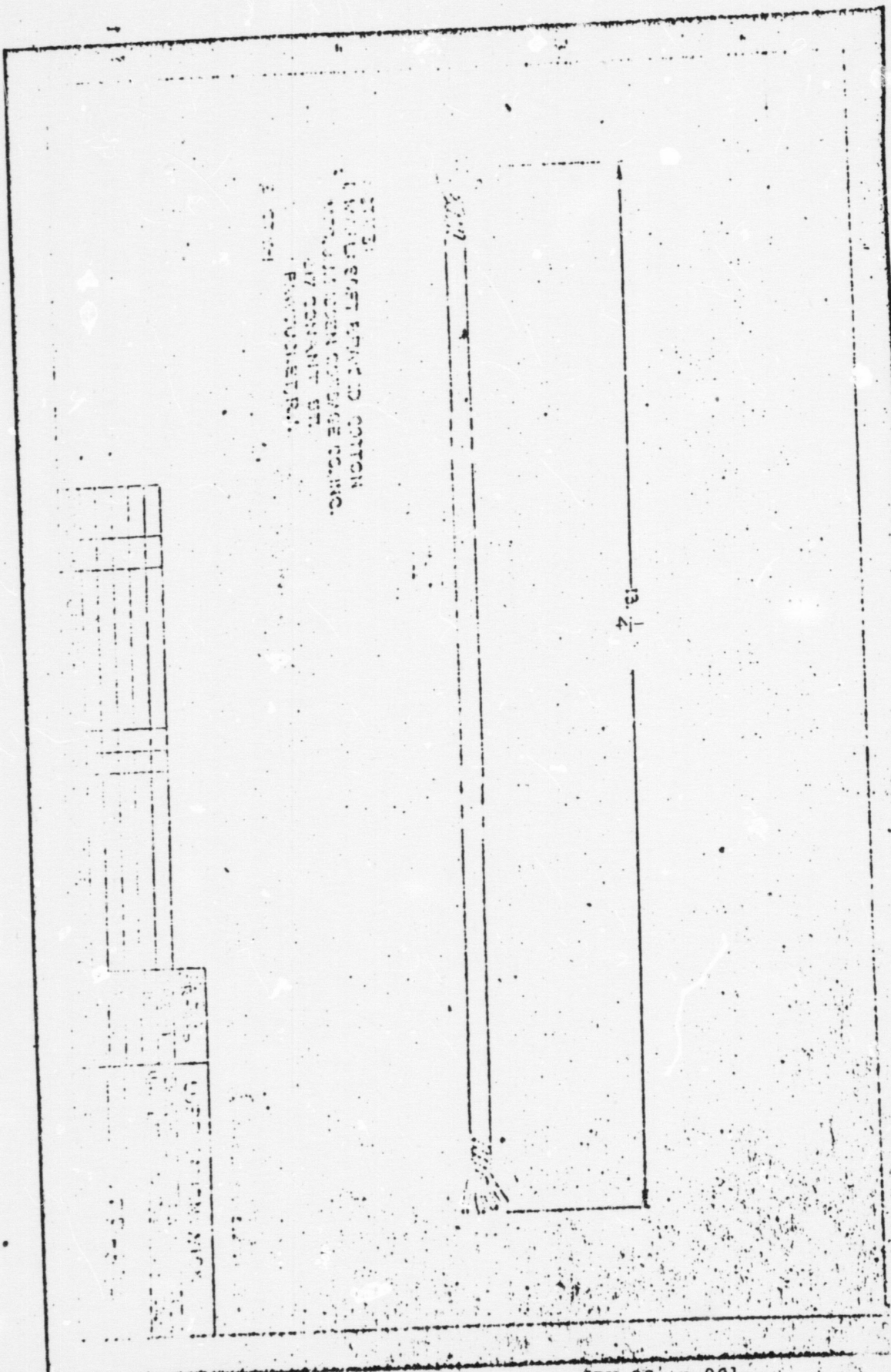


[PX 23, p 92]


FLAT
7 DIA







[PX 23, p 97]

[PX 23, p 97]

E-264

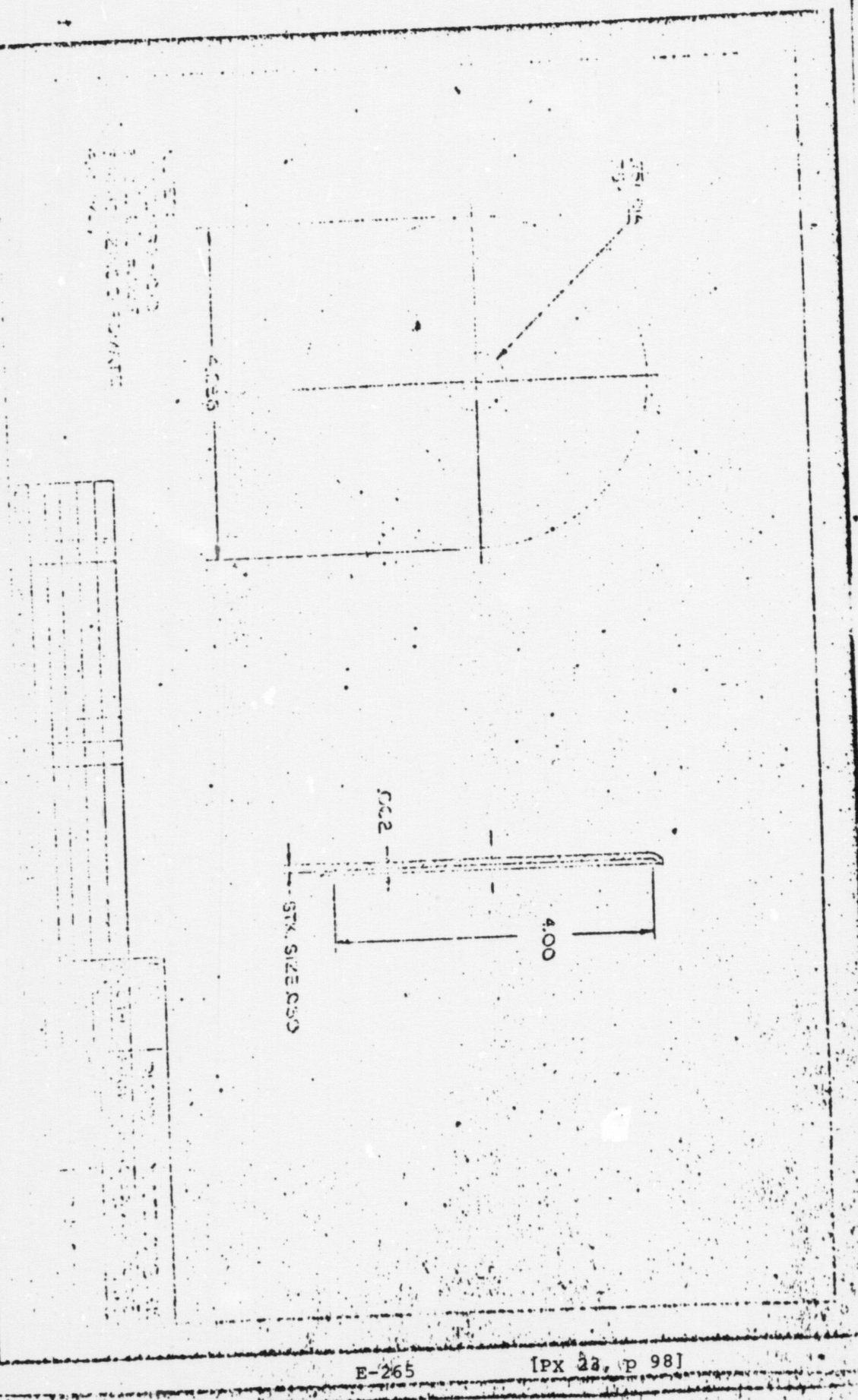
Ⓐ

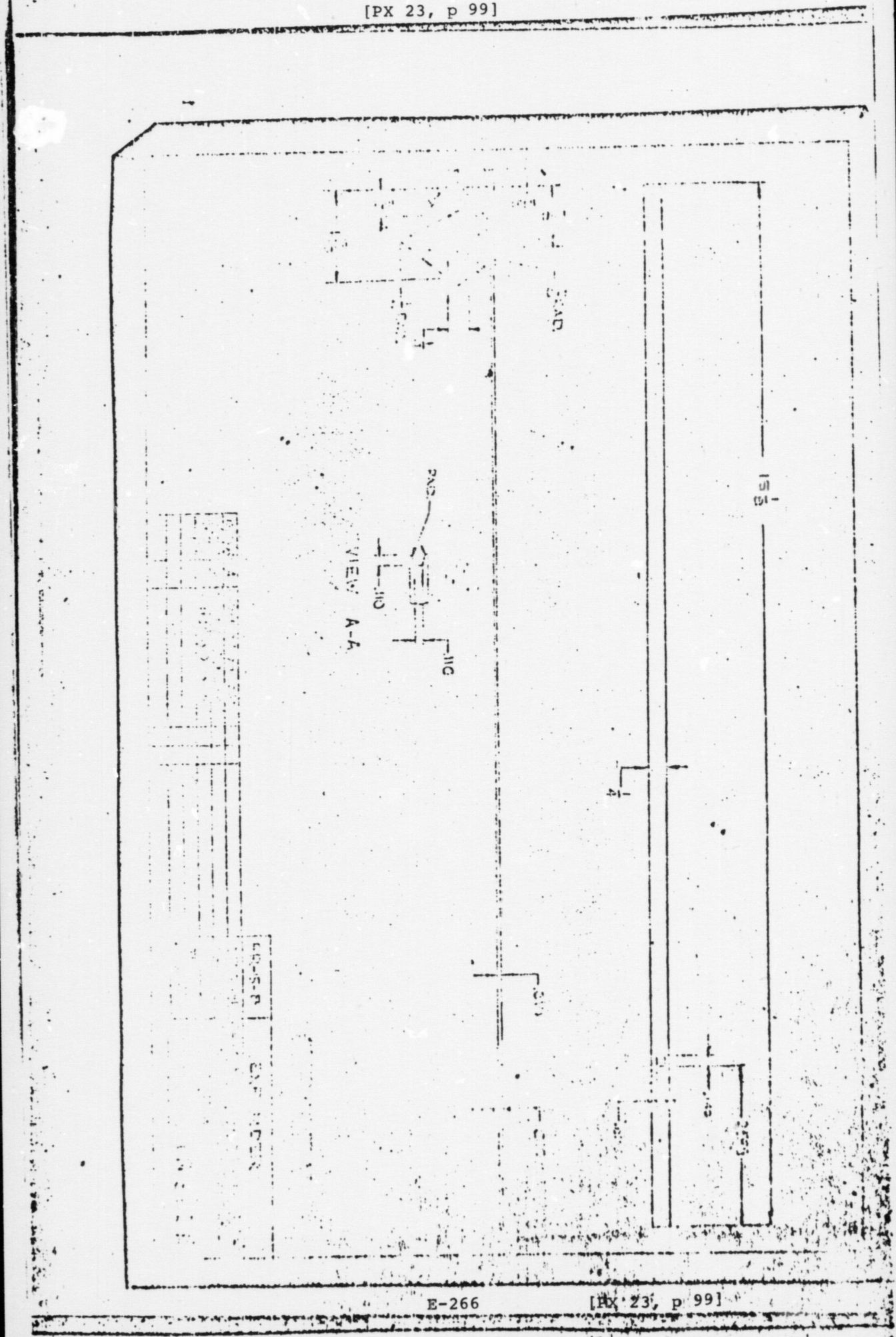
SPRING DATA:
DIAMETER OF WIRE .023
NET NO. OF COILS 6
TOTAL NO. OF COILS 8
END TYPE HELIX ORIGINAL
FREE LENGTH .500 ±.010
LENGTH COMPRESSED
LENGTH OF WIRE 318 307. +.302 -.302

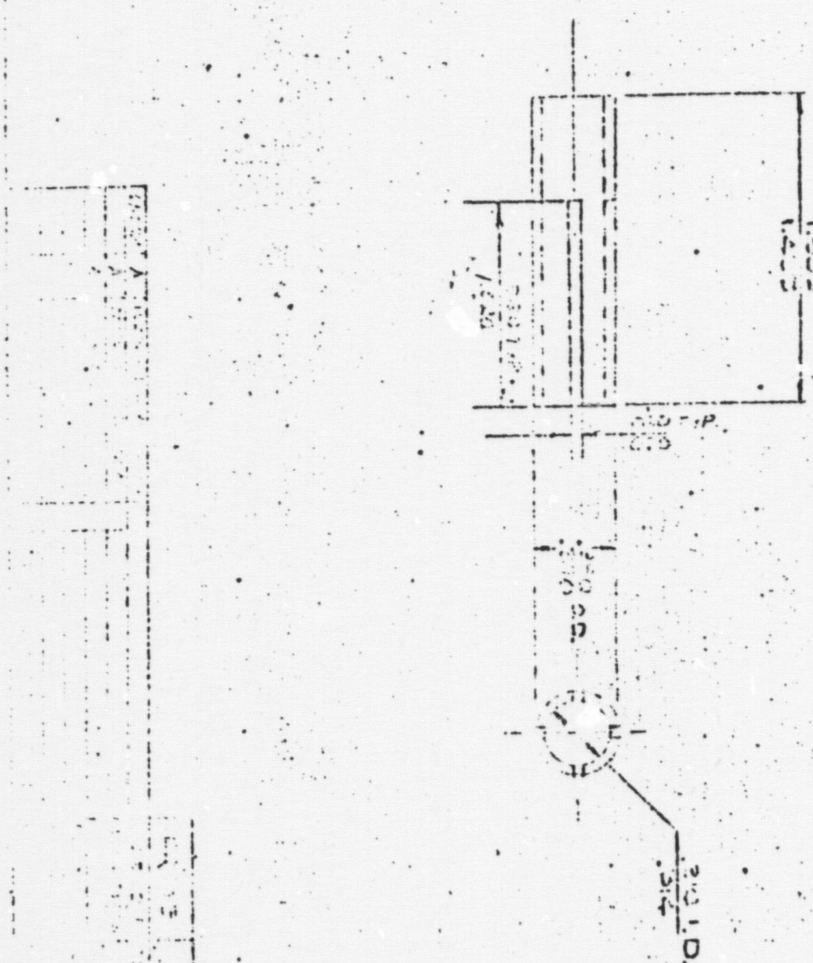
END TO BE SQUARED AND GROUND
MATERIAL MUSIC WIRE
SPECIFICATION ASTM A228

CONFIDENTIAL

FR-4-A	CHECK VALVE SPRING
	FR-[illegible]

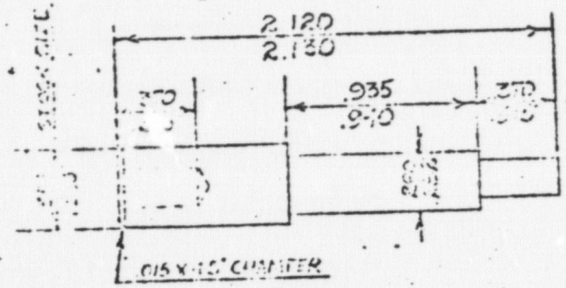

4.00
STK. SIZE C50


ROAD
PAD
VIEW A-A



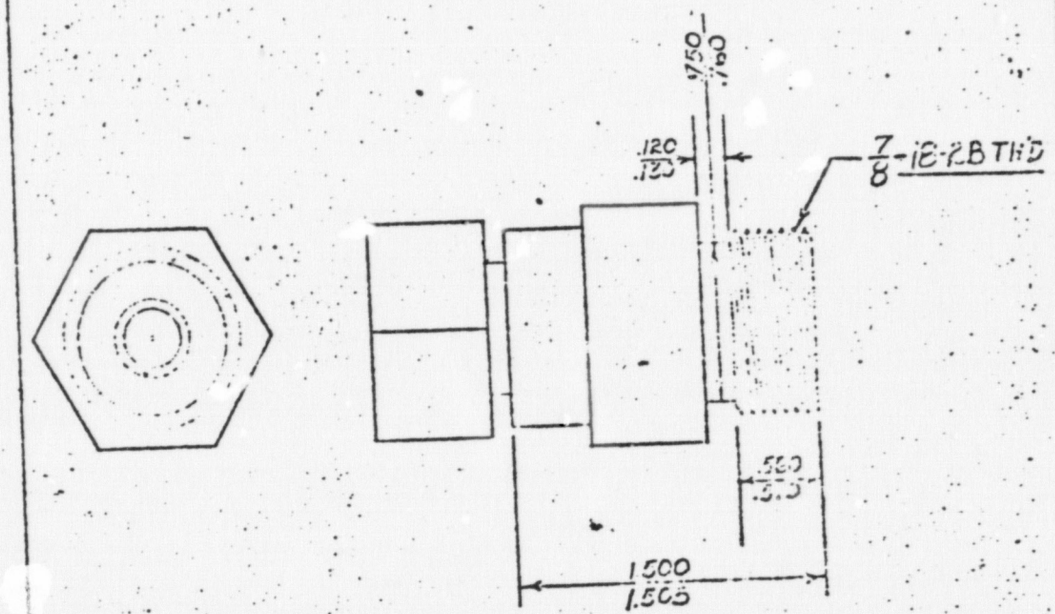
[PX 23, p 101]

SPECIAL ADAPTER SHAFT



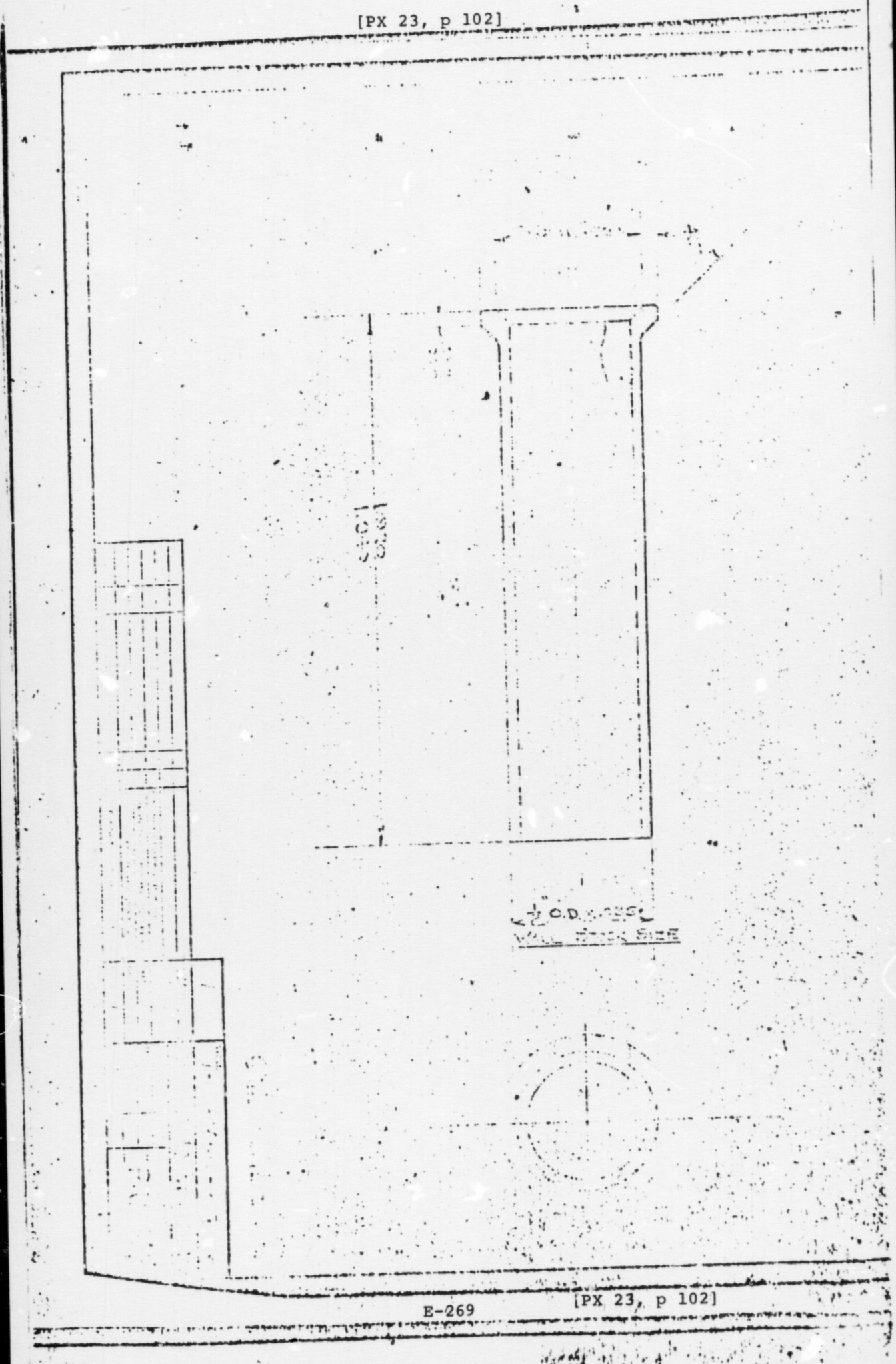
NOTE: MATERIAL- B1113

SPECIAL ADAPTER COUPLING



NOTE: MATERIAL- BLANK ADAPTER

[PX 23, p 101]
E-268

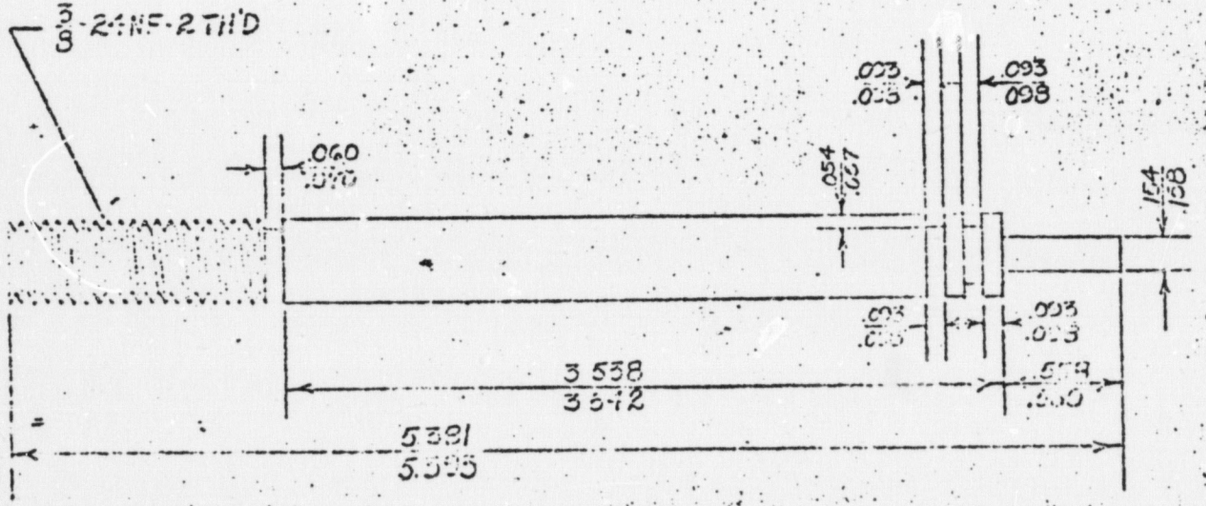


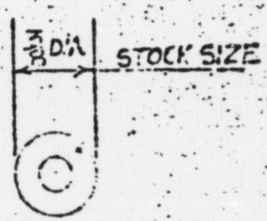
[PX 23, p 102]

[PX 23, p 103]

NOTE: MATERIAL COLD FINISHED STEEL
MACH-5

BREAK ALL CORNERS — .015 MAX.





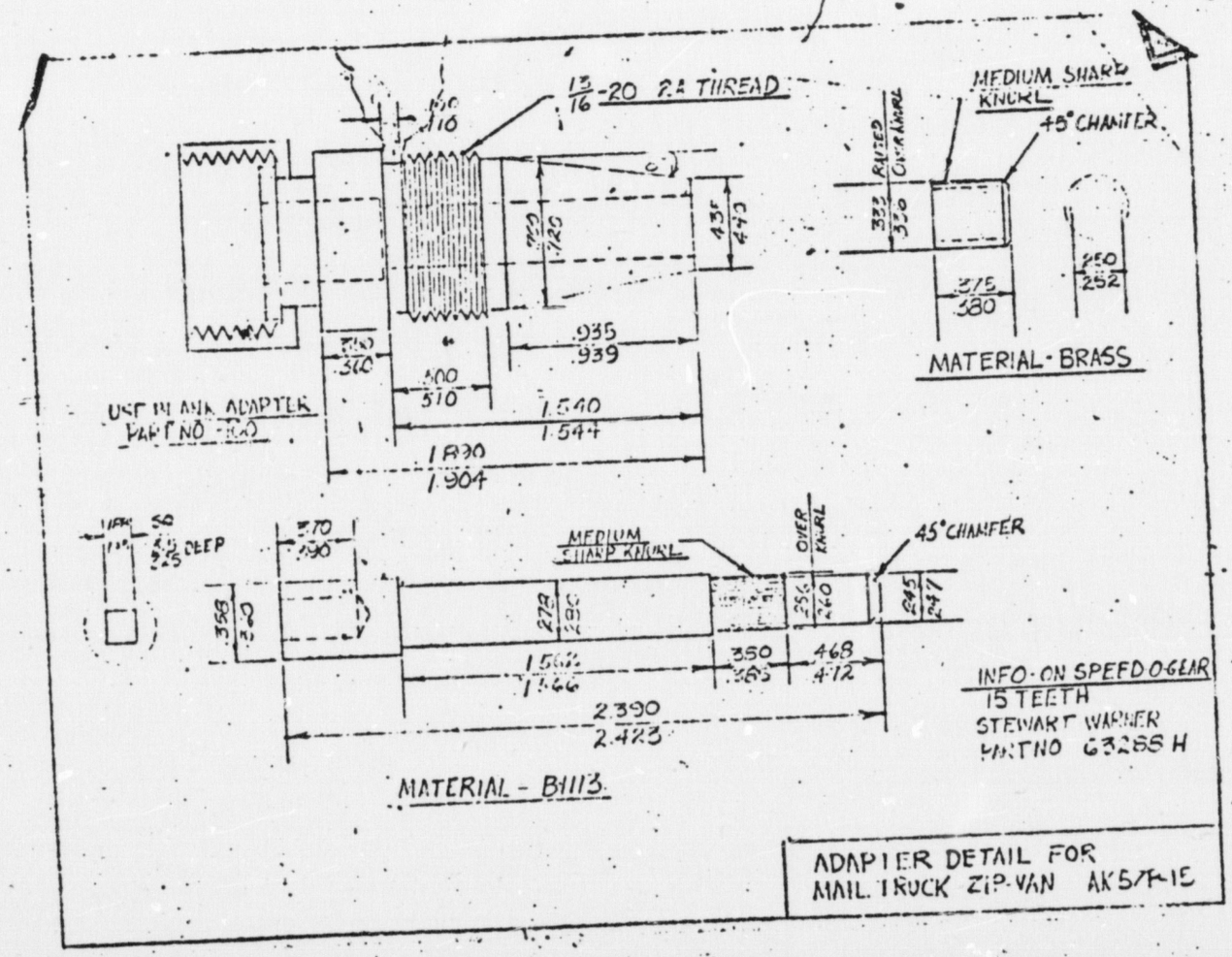
TITLE VALVE PISTON

PV25

[PX 29, p 103]

E-270

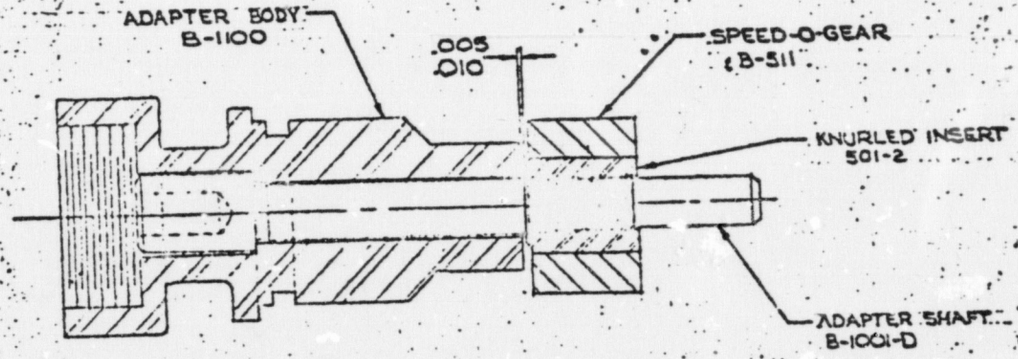
[PX 23, p 104]



[PX 23, p 104]

E-241

[PX 23, p 105]



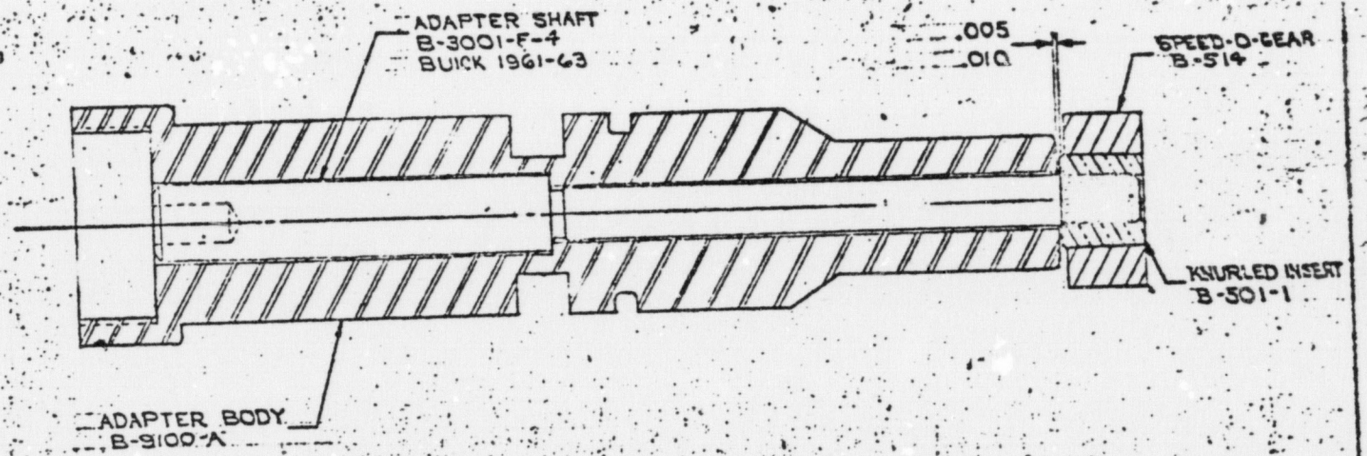
CONFIDENTIAL

						PR-36-B	FORD G4, MERC G4, FALCON AND COMET G1-G4 FAIRLANE 62,63,64
						SCALE: FULL	FORD DIVISION / DEVELOPMENT CO.
						MATL	LOCATED ON ATTACHED MISC.
						PART	POSITION / QTY.
						DWG.	AK-3FA
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQD.	

[PX 23, p 105]

E-272

[PX 23, p 106]

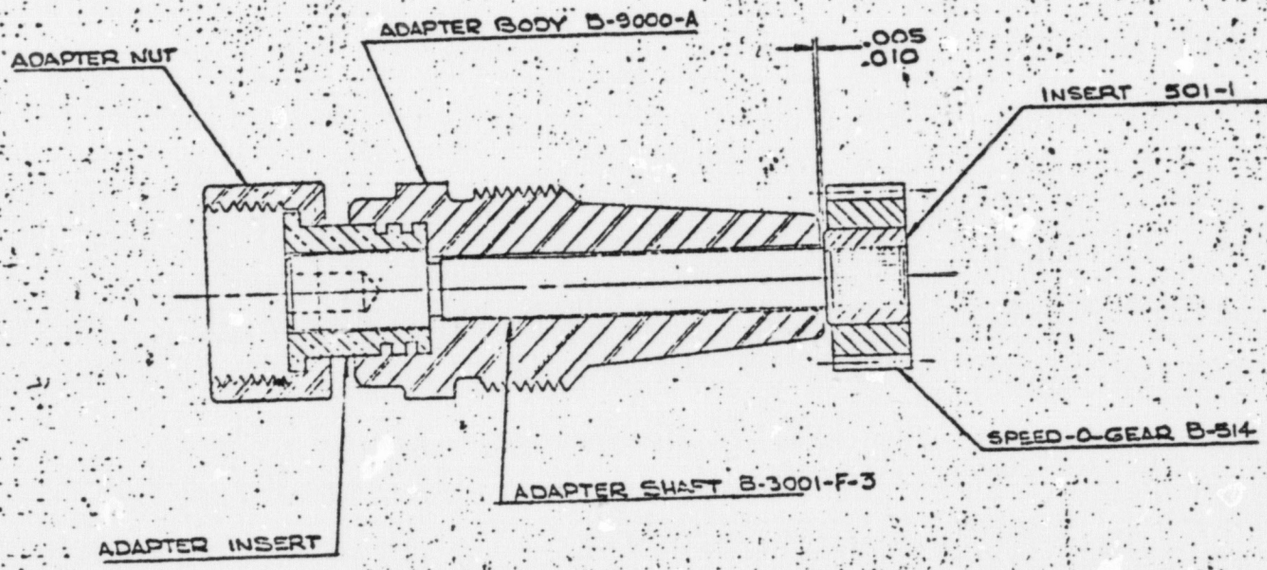


CONFIDENTIAL

						PR-39-B	BUICK 61-63
						SCALE 2:1	
						NAME	
						DATE	
						DWG.	AK-4B
DATE	LTR	CHANGE RECORD	SY	CK	DO NOT SCALE DIMENSIONS	REQD	

[PX 23, p 106]
E-273

[PX 23, p 107]



USE BLANK ADAPTER

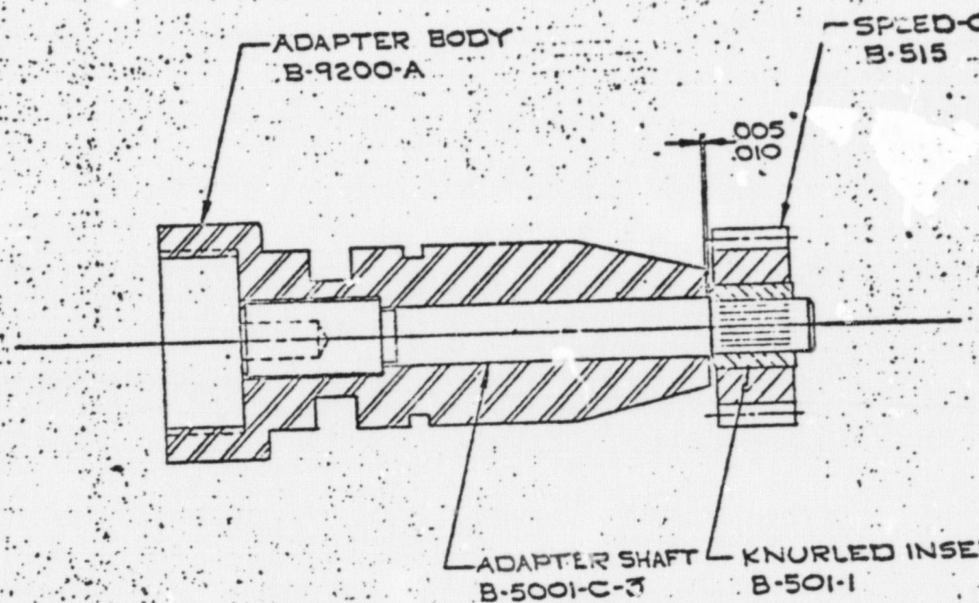
CONFIDENTIAL

	PR-37-B	TITLE BUICK 59-60
	SCALE 2-1	PERMA-FLEXIBLE CORPORATION
		DATE 1-10-61 ISSUED BY
		DRN. A.M. CHK.
		APP'D.
DO NOT SCALE DIMENSIONS		DRAWING NO. AK-4BA

[PX 23, p 107]

E-274

[PX 23, p 108]



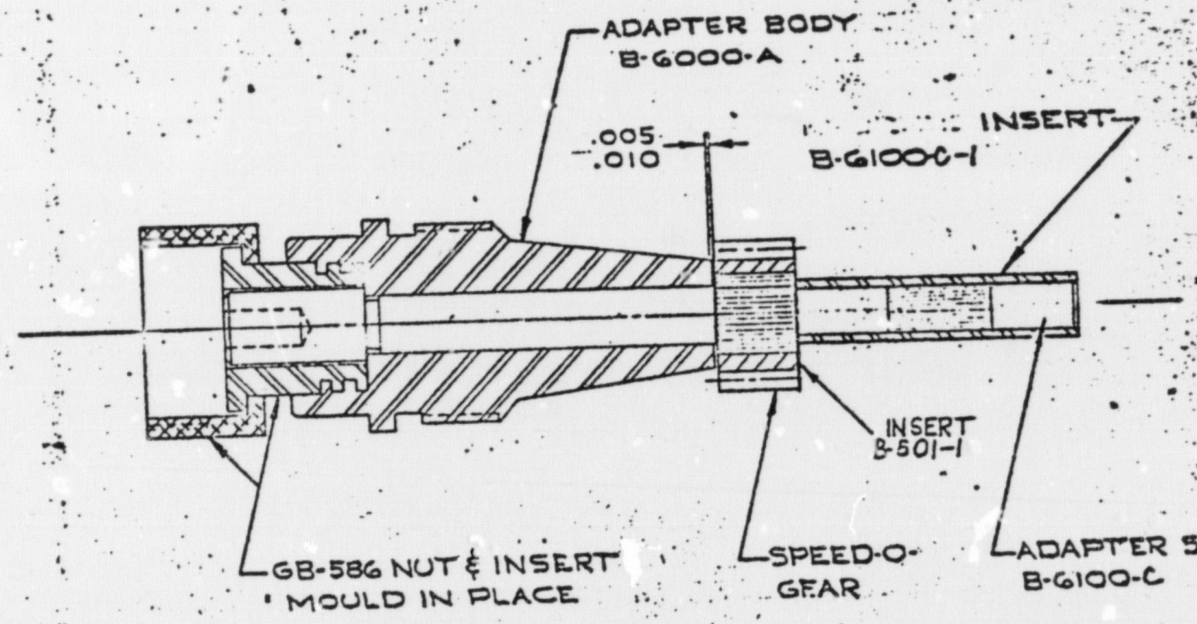
CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS

PR-26-B BUICK SPECIAL 1961/1963
SHT 2-1
PONTIAC DIVISION DEVELOPMENT CO
GENERAL MOTORS
ENGINEERING N/A
AK-4BS

[PX 03, p 108]
B-275

[PX 23, p 109]



USE BLANK ADAPTER

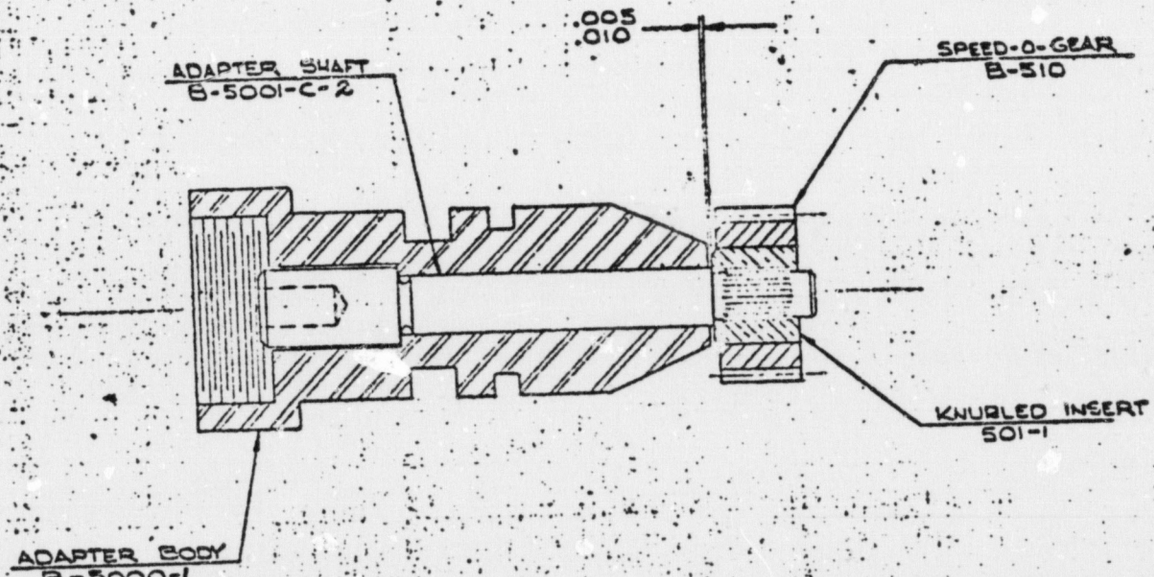
CONFIDENTIAL

						PR-27-B	CADILLAC 59-6
						SCALE 2-1	PERMA RESEARCH & DEVELOPMENT CO U.S. RTE 1 NO ATTLEBORO, MASS
							REFERENCE / / / /
DRAWN	C	ASSEMBLY CHANGE					AK-4CA
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS		

[PX 23, p 109]

E-276

[PX 23, p 110]



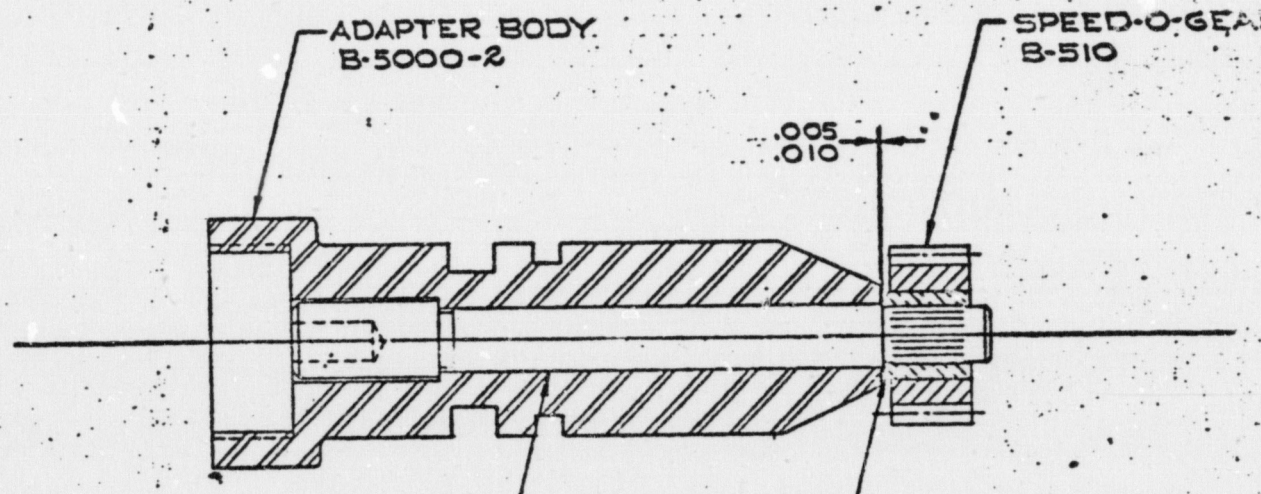
CONFIDENTIAL

								PR-29-B	TIT-2 CHEVROLET 1961-1964
								SCALE 2x1	PERINI RESEARCH & DEVELOPMENT CO. U.S. RTE 1 NORTH READING, MASS.
								MAT'L	A. MACEDO 2-14-63 NUMBER AK-4CH
								PATT	
								FINISH	
DATE	LTR	CHANGE RECORD	BY	CHK'D	DO NOT SCALE DIMENSIONS			REQ'D.	

[PX 23, p 110]

E-277

[PX 23, p 111]



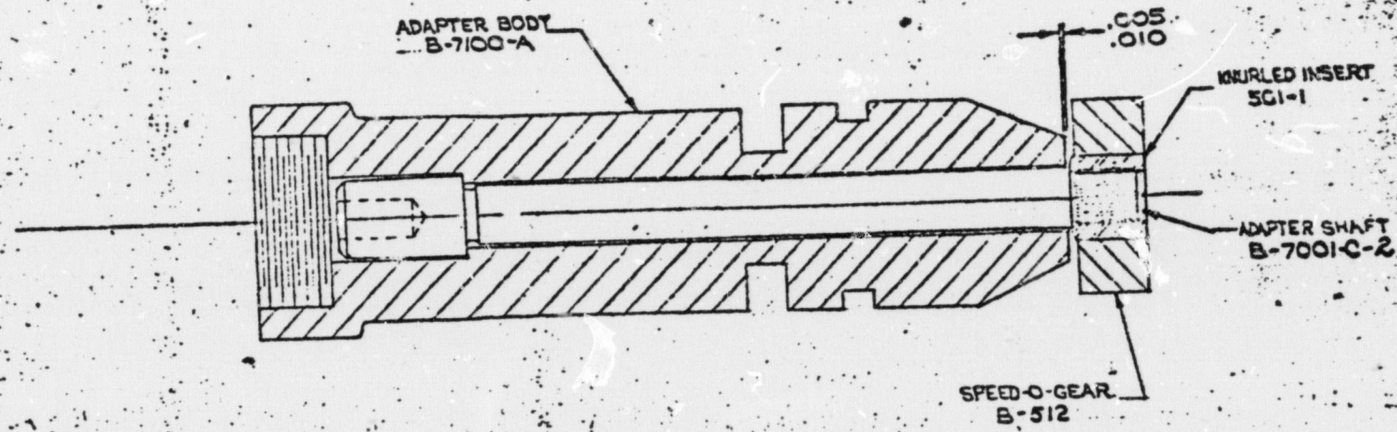
CONFIDENTIAL

						PR-30-B	CHEVROLET-1959-1960		
						SCALE 2-1	PERMA RESEARCH & DEVELOPMENT CO. U.S. RTE 1 NO. ATTLEBORO MASS.		
							REFRENCE	DWG	AK-4CHA
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS				

[PX 23, p 111]

E-278

[PX 23, p 112]

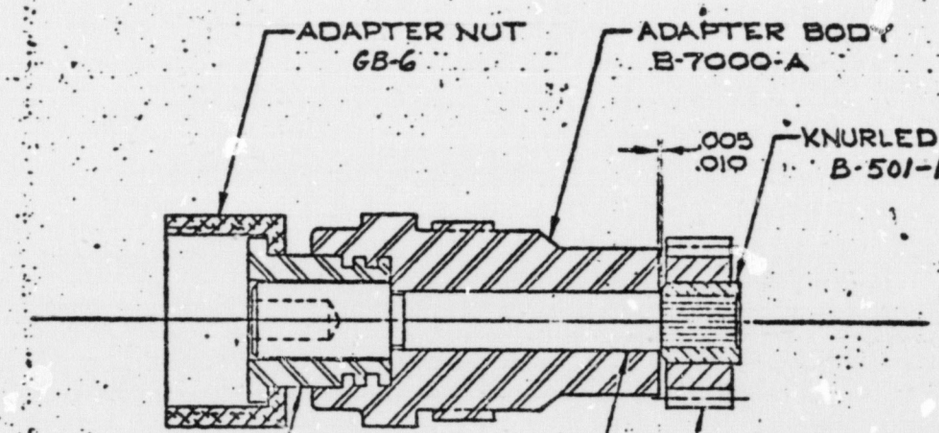


CONFIDENTIAL

										PR-35-B	PONTIAC 61-64 DELUXE
										SCALE 2x1	FERMA RESEARCH DEVELOPMENT U.S. RTE 1 NO ATTLEBORO MASS.
										MAT'L	APPROVED 10-30-63
										PATT.	AK-4P
										FIN.	
DATE	L.R.	CHANGE RECORD				BY	CK.	DO NOT SCALE DIMENSIONS		REQ'D	

[PX 23, p 112]

[PX 23, p 113]



USE BLANK ADAPTER

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CL

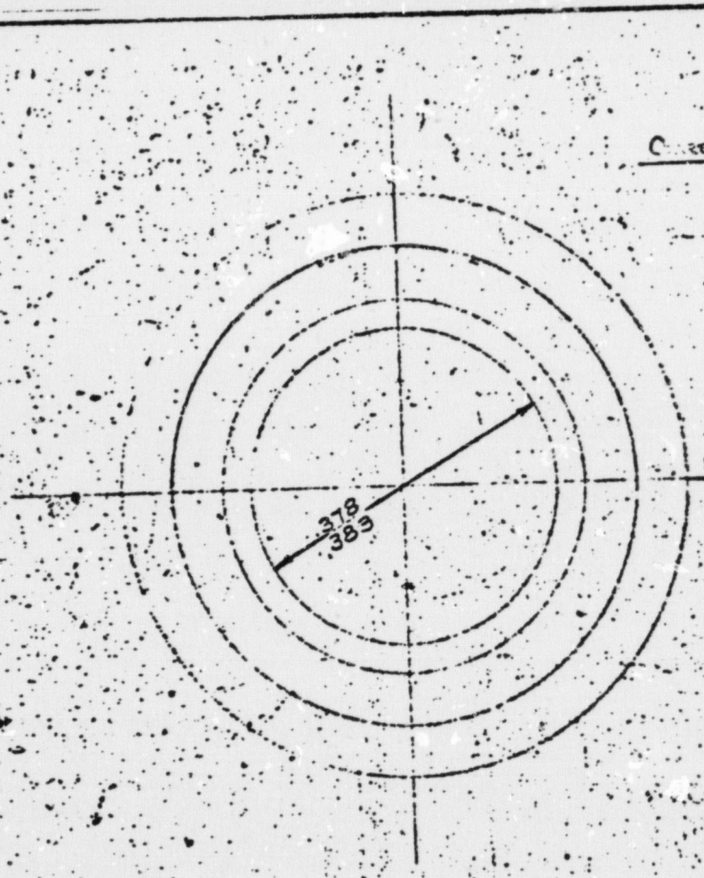
DO NOT SCALE DIMENSION

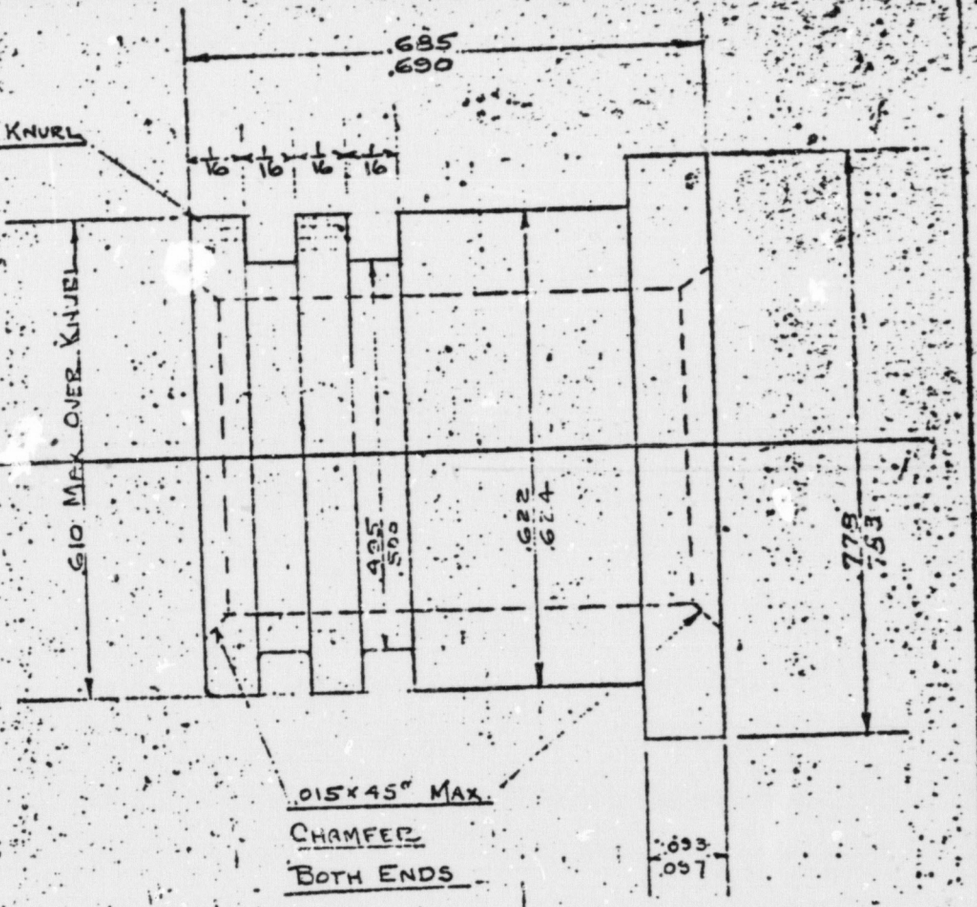
PR-25-B	PONTIAC GO-GA 4 SPEED
SCALE 2-1	FERMA[illegible] DEVELOPMENT CO. U.S. RT. [illegible] ATTLEBORO, MASS.
[illegible]	AK-4PA

[PX 25, p 113]

E-280

[PX 23, p 114]





NOTE:
CADMIUM OR ZINC
PLATE .0002 MIN
DIMENSIONS APPLY AFTER PLATING

CONFIDENTIAL

					SCALE 2/1	TITLE CONNECTION NUT SLE...
					WEIGHT	
				UNLESS OTHERWISE INSTRUCTED, BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	DATE	DRAWING NO. GB-5
					DRAWN	
12-17-[illegible]	1/1	[illegible]			CHK'D	
DATE	SER.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	APPD

[PX 20, p 114]
E-281

[PX 23, P 115]

CONFIDENTIAL

TITLE: Blank Adapter Connection Nut

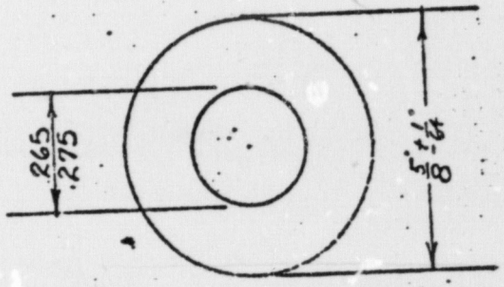
DRAWING NO. GB-6

MATERIAL CHANGED

DO NOT SCALE DIMENSIONS

E-282 [PX 23, P 115]

[PX 23, p 116]





NOTE: MATERIAL: OPTIONAL
STEEL, NYLON OR BRASS

CONFIDENTIAL

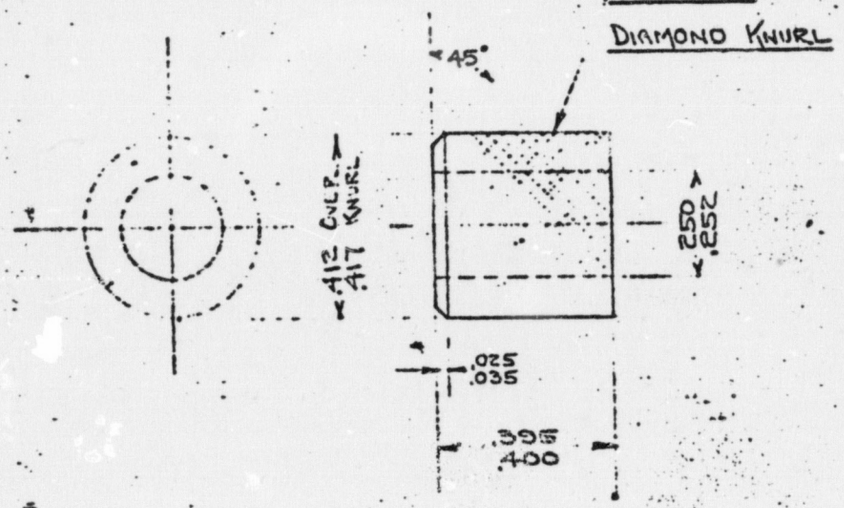
	TITLE WASHER	
SCALE 4x1	PERMA RESEARCH & DEVELOPMENT CO US RTE 1 N ATTLEBORO, MASS.	
MAT'L SEE NOTE	DR BY RDM	11/21/61
FATT	CK BY	
FINISH	ENG BY	
REQ'D	APPVD	

PR175

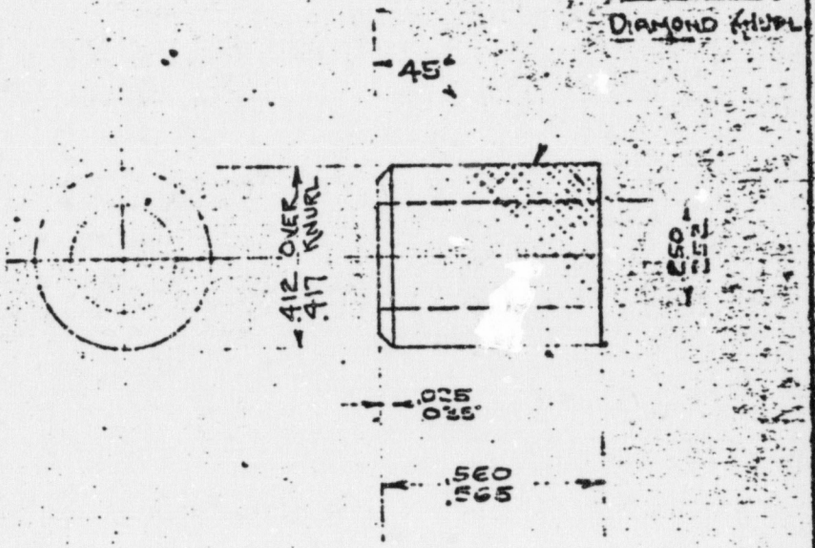
[PX 23, p 116]

E-283

[PX 23, p 117]



501-1



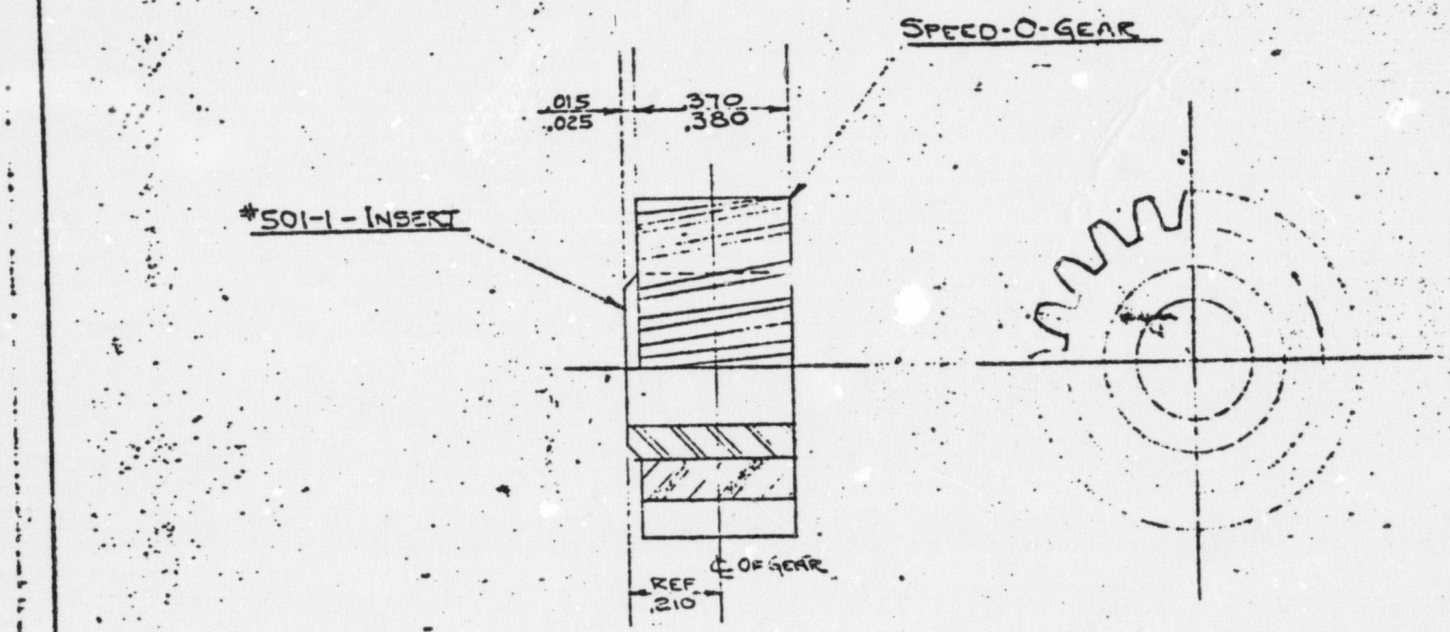
501-2

						SCALE 4 X 1	TITLE SPEED-O GEAR INSERTS	
						MATERIAL	TERMA RESEARCH & DEVELOPMENT CO. L. S. RTE. 2 NO. ATTLEBORO MASS	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	FINISH CUTTING	DR BY DAW	DATE
					BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PATT.	CH BY	DRAWING NO B-501
					± TOLERANCE ON FRACTIONAL DIMENSIONS	FINISH	ENG BY PETT	DATE
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D	APPROV	DATE

[PX 23, p 117]

E-284

[PX 23, p 118]



FORD & MERCURY THRU 1963

17T LT BLUE	18T PURPLE	19T GRAY	20T BLACK	21T YELLOW
FORD PART NO C5AA-17271A	FORD PART NO C0AA-17271B	FORD PART NO C0AA-17271C	FORD PART NO C0AA-17271D	FORD PART NO C0AA-17271E
PERMA PART NO AK3F-17	PERMA PART NO AK3FF-15	PERMA PART NO AK3FF-19	PERMA PART NO AK3FF-20	PERMA PART NO AK3FF-21

CHRYSLER - 1962-1963

17T ORANGE	18T DARK PURPLE	19T DARK BLUE
CHRYSLER PART NO 2204356	CHRYSLER PART NO 2204357	CHRYSLER PART NO 2204358
PERMA PART NO AK2A-17	PERMA PART NO AK2A-18	PERMA PART NO AK2A-19

CHEVROLET 1961-1963

18T BROWN	20T BLUE
CHEVROLET PART NO 3777063	CHEVROLET PART NO
PERMA PART NO AK4CH-18	PERMA PART NO AK4CH-20

FORD & MERCURY THRU 1963

17T LT BLUE	18T NATURAL	19T RED	20T ORANGE
FORD PART NO C0AA-17271A	FORD PART NO C0AA-17271-J	FORD PART NO C0AA-17271-K	FORD PART NO C0AA-17271-E
PERMA PART NO AK3F-17	PERMA PART NO AK3F-18	PERMA PART NO AK3F-19	PERMA PART NO AK3F-20

CONFIDENTIAL

REV	LTR	CHANGE RECORD	BY	CK
1/15/64	A	CHANGED WIDTH OF SLOT FROM .370/.380 TO .375/.385 AND NEW SIZE GIVEN.	JJ	

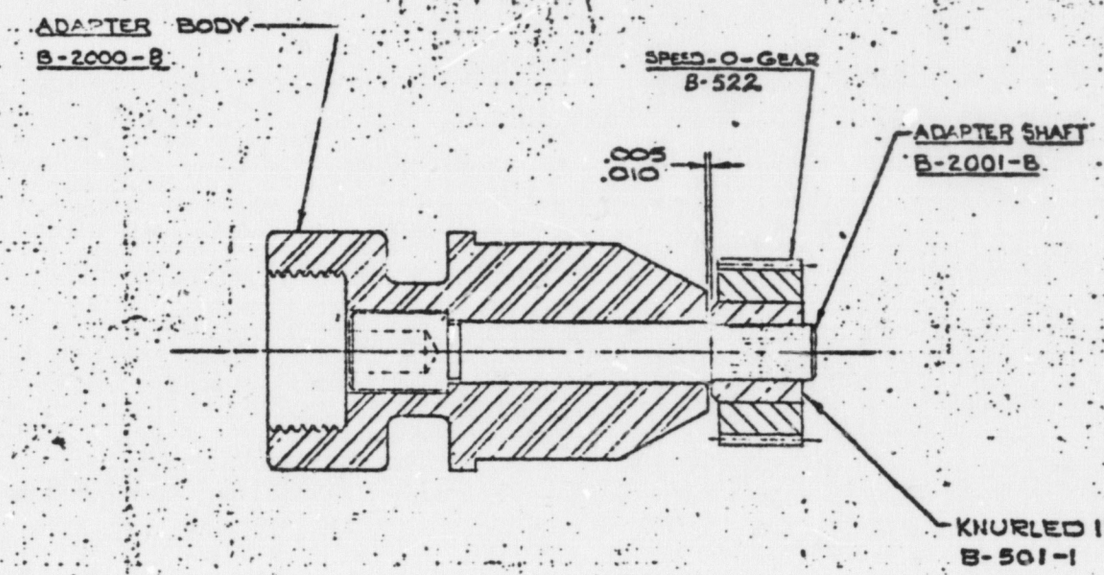
FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED:
BREAK ALL SHARP CORNERS AND REMOVE ALL BURR
X ALLOWANCE ON FRACTIONAL DIMENSIONS.
DO NOT SCALE DIMENSIONS

SCALE 4 FULL
MATL
FINISH
REQ'D

TITLE: NASHUA SPEED-O-GEAR CASE
PERMA MFG. Co. & SUBSIDIARIES
DRAWING NO. B-510A

[PX 23, p 118]
E-285

[PX 23, p 119]



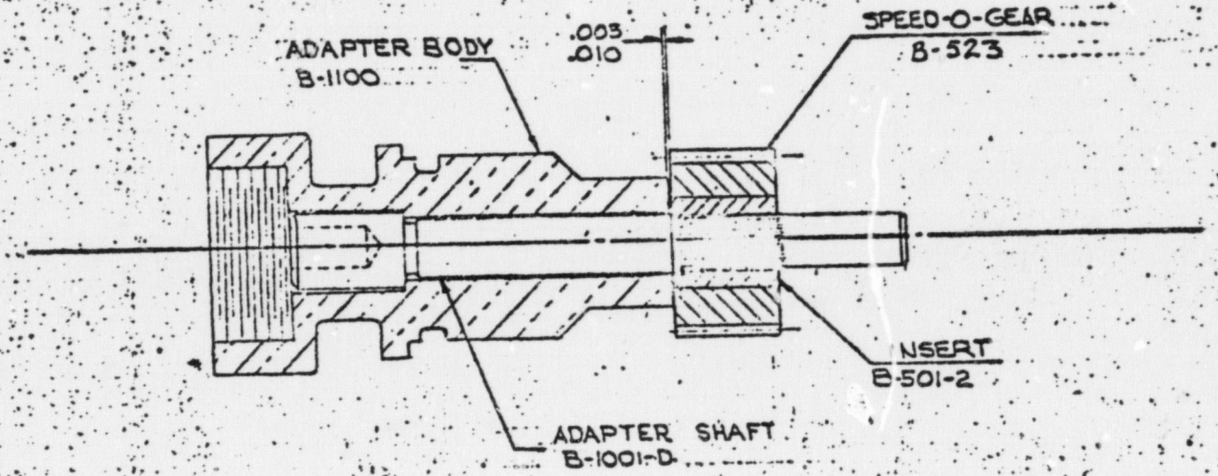
CONFIDENTIAL

	PR-31-B	TITLE RAMBLER 59-62	
	SCALE 2/1	PERMA RESEARCH & DEVELOPMENT CO. US RTE 1 NO. ATTLEBORO MASS.	
	MATL.	DR BY E.N.G. DATE 1-5-59	DRAWING NO. AK-IRA
	FINISH	TR BY	
DO NOT SCALE DIMENSIONS	CLASS	CHK BY	
		APPVD	

[PX 23, p 119]

E-286

[PX 23, p 120]

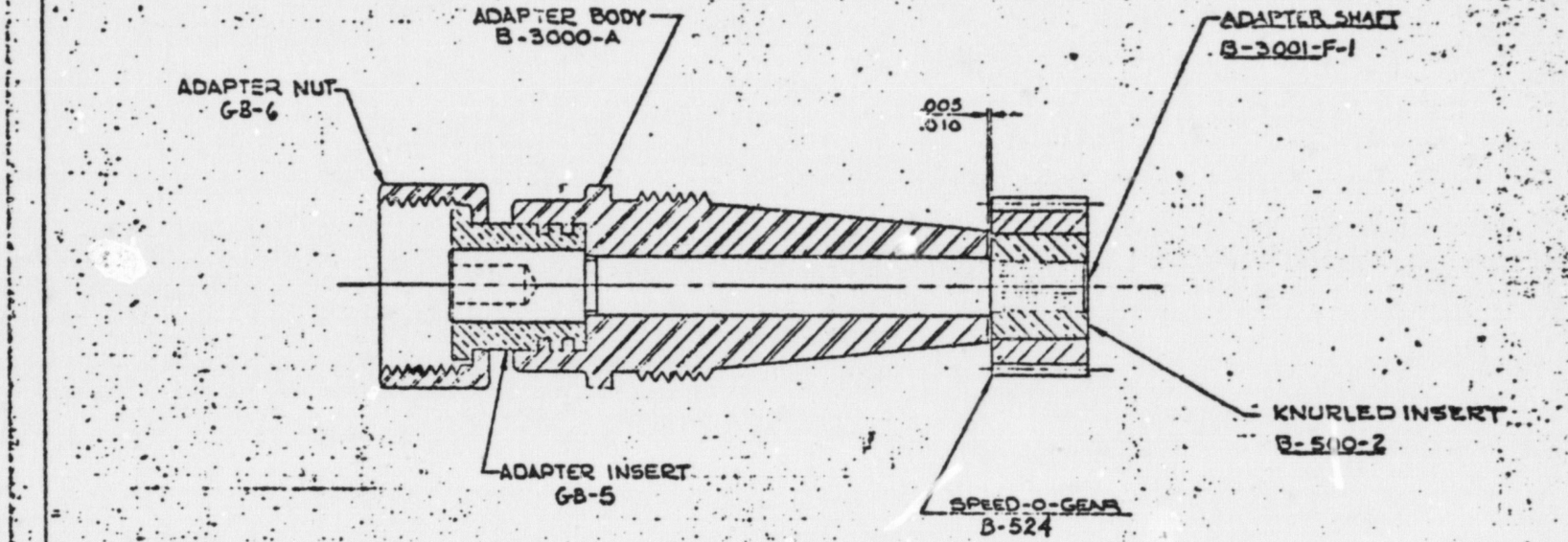


CONFIDENTIAL

						PR-38-B	RAMBLER 1963-1964	
						SCALE 2X1	PERMA RESEARCH & DEVELOPMENT CO.	
						MATL	U.S. RTE. 1 NO. ATTLEBORO, MASS.	
							POSITION	DWG.
						MATT.		AK-1R
						FIN		
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D.		

[PX 23, p 120]

[PX 23, p 121]



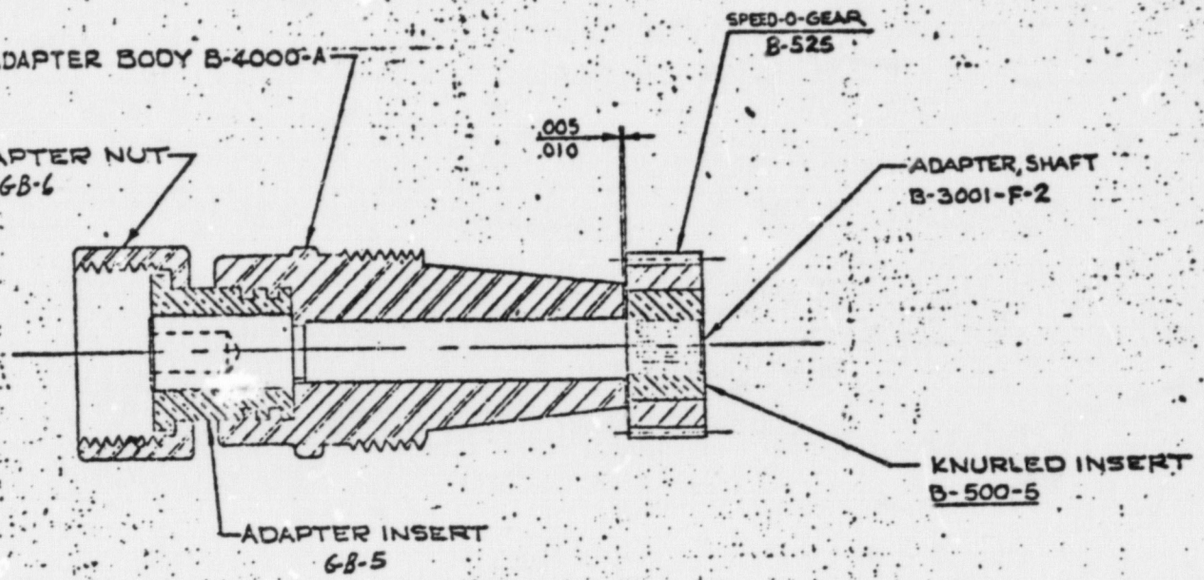
USE BLANK ADAPTER

CONFIDENTIAL

	PB-32-B	FOR CHRYSLER PRODUCTS-V8 POWER-FLYTL, 59-61
	SCALE 2/1	PLANA RESEARCH & DEVELOPMENT CO. U.S. RTE 1, NO. ATTLEBORO, MASS.
		DRAWING NO. AK-2AP
DO NOT SCALE DIMENSIONS		

[PX 23, p 121]

E-288

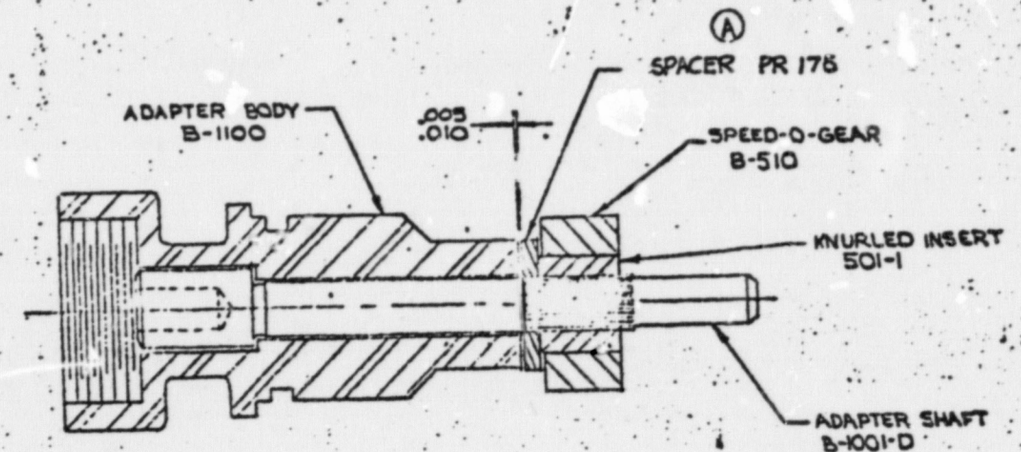


USE BLANK ADAPTER

CONFIDENTIAL

	P2-33-B	FOR CHRYSLER PRODUCTS-V8 "TORQUE FLYTE" 55-61
	SCALE 2/1	PERMA RESEARCH & DEVELOPMENT CO. U.S. RTE 1, NO ATTLEBORO MASS.
	MATL	
	PATT.	DRAWN EVG / TRACED / CHKD / ENGR / APPD / DRAWING NO. AK-2AT
DO NOT SCALE DIMENSIONS	ALLOW	

[PX 23, p 123]

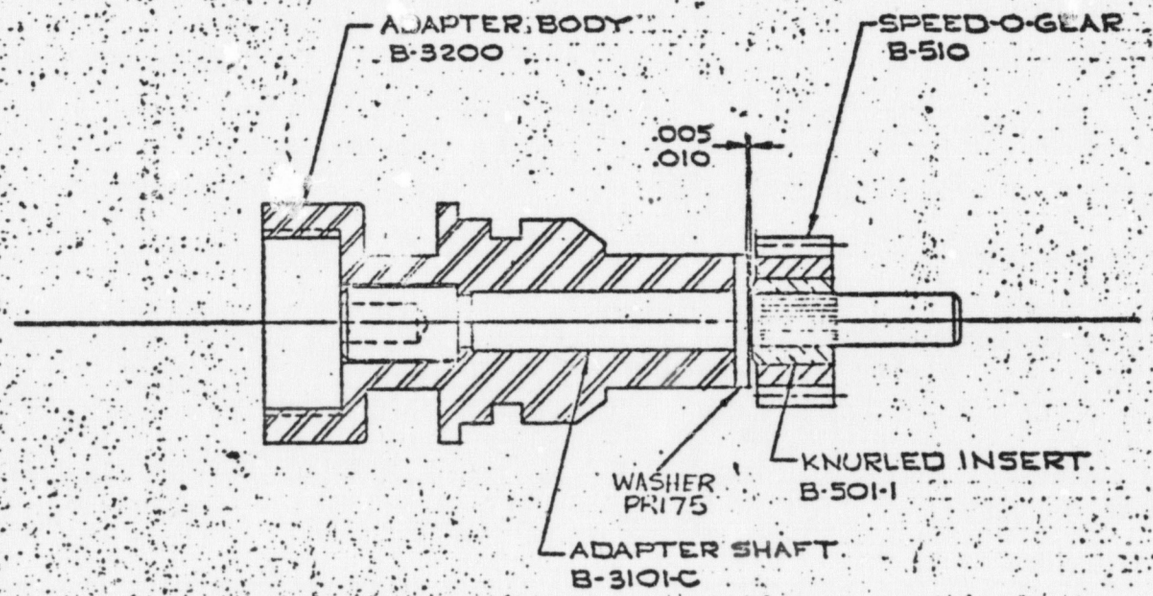


CONFIDENTIAL

						PR-34-B	FORD MERC. T-BIRD 59-63
						SCALE 2:1	PERMA RESEARCH & DEVELOPMENT CO.
						MAT'L	U.S. RTE. 1 NO. ATTLEBORO MASS.
							R POISSON
						PATT	
	A	SPACER PRTS ADDED				F.IN	AK-3F
DATE	ER	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQD	

E-290 [PX 23, p 123]

[PX 23, p 124]



CONFIDENTIAL

						PR-28-B	CHRYSLER PRODUCTS 62-64 V8, 6 CYL 63-64 6 CYL
						SCALE 2-1	PERMANENT FUEL INJECTION DEVELOPMENT CO. UNITED TECHNOLOGY CENTER
							NEXT ASSY.
6/15/64	A	ADD WASHER PR175					AK-2A-A
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DRAWING		

[PX 23, p 124]

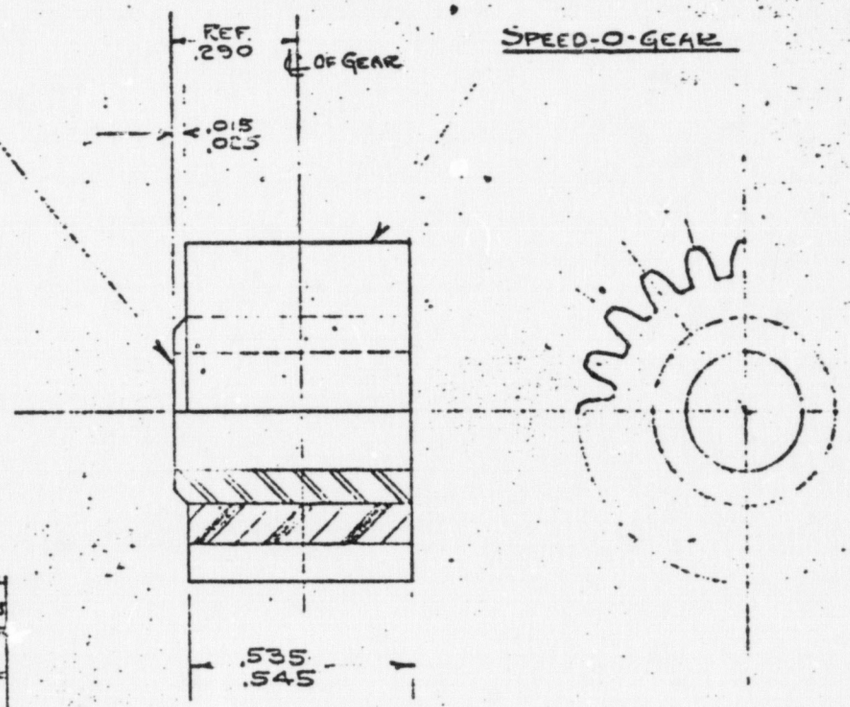
[PX 23, p 125]

[PX 23, p 125]

E-292

SPEED-O-GEAR

501-2 INSERT



FORD-1364--FALCON-COMET-61-63--FAIRLANE 500-62-63

17T	18T	19T	20T	21
GREEN	GOLD	PINK	BLACK	RED
FORD PART NO C3DA-1271-B	FORD PART NO C3DZ-1271-B	FORD PART NO C3DD-1271-C	FORD PART NO C1DD-1271-A	FORD PART NO C4DA-1271-A
PERMA PART NO AK3EA-17	PERMA PART NO AK3EA-18	PERMA PART NO AK3EA-19	PERMA PART NO AK3EA-20	PERMA PART NO AK3EA-21

DATE	LTR	CHANGE RECORD	BY	CK

DO NOT SCALE DIMENSIONS

SCALE 4/1

MATERIAL STEEL

BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS

± ½ ALLOWANCE ON FRACTIONAL DIMENSIONS

PART

PINION

REQ'D

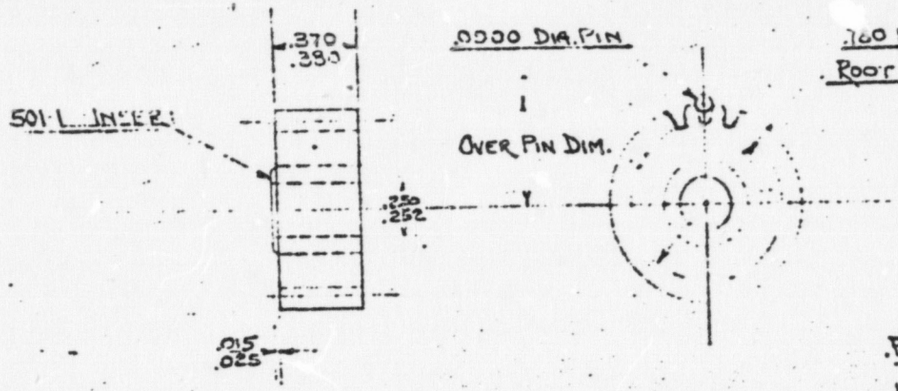
TITLE WIDE SPEED-O-GEAR P.C.D.

PERMA RESEARCH & DEVELOPMENT CO.

DRAWING NO B-511

[PX 23, p 126]

[PX 23, p 126]
E-293



NORMAL DIAMETRAL PITCH — 26

NORMAL PRESSURE ANGLE — 22.500°

BACKLASH WHEN CHECKED WITH MASTER MATING GEAR ON 12.175 CENTER DISTANCE — .004 .000

.PITCH DIAMETER OF GEAR MUST BE CONCENTRIC WITH .250 / .252 DIA WITHIN .004 T.I.R.

GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

HELICAL GEAR TOOTH DATA

PONTIAC PART NO	PERMA PART NO	NUMBER OF TEETH	IDENTIFICATION	HELIX ANGLE L.H.	PITCH CIRCLE DIA	LEAD	OUTSIDE DIAMETER	OVER PIN DIM. REF.
543942	AK4P-19	19	DARK GRAY	13.2264°	.7507	10.0340	.920 / .926	.4942 / .4970
543944	AK4P-21	21	ORANGE	12.8946°	.8320	10.5666	.942 / .948	.5095 / .5127

DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQ'D

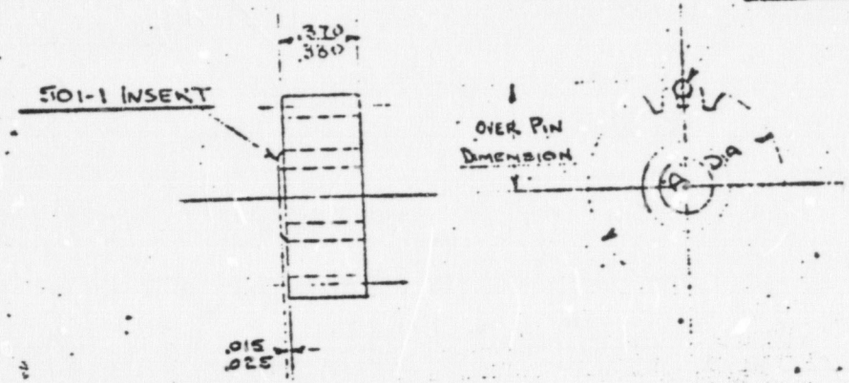
FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED
BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS
ALLOWANCE ON FRACTIONAL DIMENSIONS

SCALE 2 X 1
MATL STEEL
FINISH 101
DRAWN BY N.N. DATE 5-12
TR BY
CHK BY DATE
APP'D DATE

TITLE SPEED-O-GEAR FOR PONTIAC
PERMA RESEARCH & DEVELOPMENT CO.
N. W. 119TH ST. MIAMI, FLORIDA
DRAWING NO. B-512

[PX 23, p 127]

[PX 23, p 127]



NOTE:

NORMAL DIAMETRAL PITCH -- 26

NORMAL PRESSURE ANGLE -- 25°

HELIX ANGLE R.H. SPECIAL -- 8°-3'-33"

GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

BUICK PART NO.	PERMA PART NO	NO OF TEETH	IDENTIFICATION	PITCH DIAMETER	LEAD	FULL DEPTH	BACKLASH	REFERENCE DIMENSION OVER PIN	"A" DIA
1196184	AK-48-19	19	BLUE	.738036	16.3811	.086/.090	.004/.008	.878--.880	.850-.855
1196192	AK-48-20	20	RED	.77688	17.24332	.086/.090	.004/.008	.888--.890	.850-.855
1196193	AK-48-21	21	GREEN	.815 24	18.105486	.086/.090	.004/.008	.889--.891	.850-.855

1963

TITLE: PINION - SPECIAL DRIVE

SCALE 2 X 1

DATE 2-7-63

PERMA RESEARCH & DEVELOPMENT CO. U.S. RTE. 1 NO. ATTLEBORO, MASS.

DRAWING NO. B-514

FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED

BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS

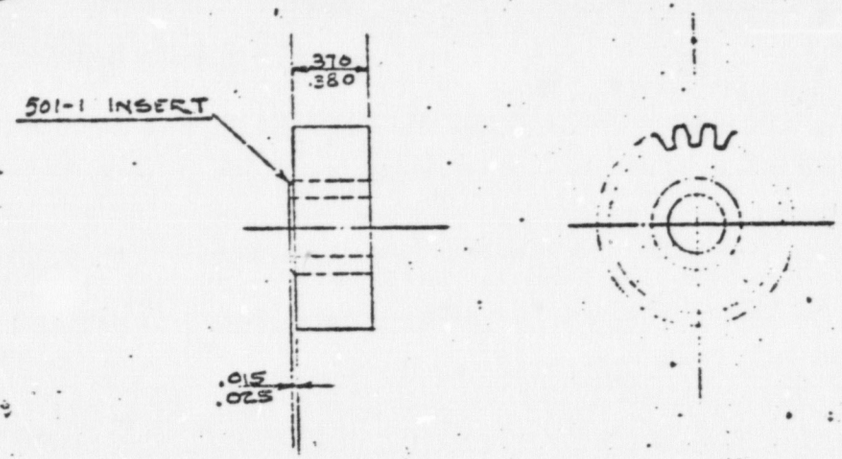
ALLOWANCE OR FRICTIONAL RELATIONS

DO NOT SCALE DIMENSIONS

DATE	LTR.	CHANGE RECORD	BY	CK.

[PX 23, p 127]

[PX 23, p 128]



NOTE:

NORMAL HELIX ANGLE -- 11°-52'- R.H.

NORMAL PRESSURE ANGLE -- 20°

NORMAL CIRCULAR PITCH (REF.) -- .1128

NORMAL DIAMETRAL PITCH -- .28

MAX. SPACING ERROR ON CIR. PITCH -- .0002

MAX. LEAD ERROR -- .0005

INVOLUTE PROFILE ERROR -- ±.0003

BACKLASH WHEN CHECKED WITH MASTER MATING GEAR ON MIN. CENTER DIST. 1.0597 TO BE .004-.008

QUICK SPECIAL PART NO	PERMA PART NO	NO. OF TEETH	IDENT COLOR	PITCH DIA REF.	LEAD	BASE CIRCLE DIAMETER	NORMAL CIRCULAR THICKNESS (REF.)	
164712	FK-485-19	19	BLUE	.6934	10.3679	.6499	.0655 -- .0674	
343453	FK-485-20	20	RED	.7299	10.9135	.6841	.0522 -- .0541	
164712	FK-485-21	21	GREEN	.7664	11.4592	.7183	.0399 -- .0409	

NOTE:

GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS

1961-1963

TITLE: BUICK SPECIAL SPEED-O-STAT

SCALE 2X1

DATE 7-7-62

-01

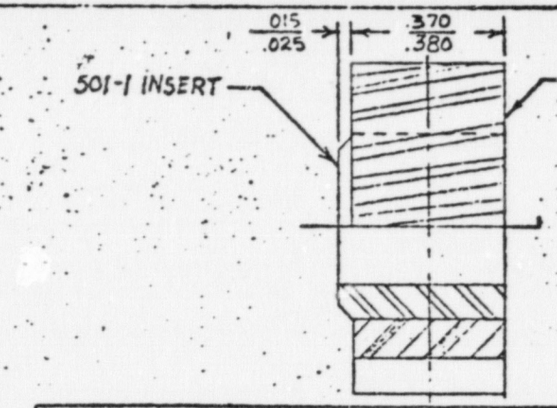
FINISH

PERMA RESEARCH AND DEVELOPMENT CO. U. S. RTE. 1, NO. ATTLEBORO, MASS.

DRAWING NO. B-515

[PX 23, p 128]
B-295

[PX 23, p 129]



NOTES
1. PITCH DIAMETER OF GEAR MUST BE CONCENTRIC WITH .250-.252 DIA. WITHIN .004 T.I.R.
2. GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

PONTIAC 1960-64
4-SPEED TRANSMISSION

HELICAL GEAR DATA (GENERATING)

PART NUMBER	PERMA NUMBER	NUMBER OF TEETH	IDENTIFICATION		LEAD	PITCH DIA.	CIRCULAR TOOTH THICKNESS (NORMAL)	OVER PIN DIM. REF	PIN DIA
5+-297	AK-4PA-18	18	18T	YELLOW	8.6960	.7150		.4354/.4329	.0835

NORMAL DIAMETRAL PITCH — 26
NORMAL PRESSURE ANGLE — 22°30'
HELIX ANGLE, R.H. — 14°29'0"
OUTSIDE DIAMETER — .812/.806
BACKLASH WHEN CHECKED WITH MATING GEAR ON 1.006 CENTER — .002-.005

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSION

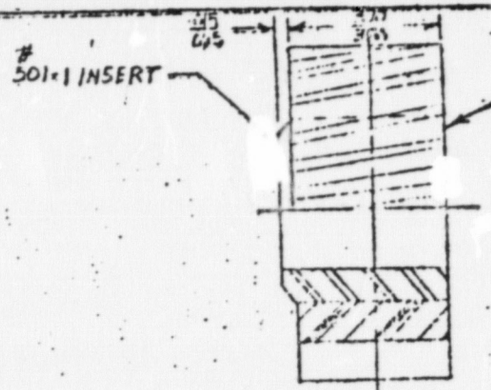
SCALE 4X1
MATL. ZYTEL
PATT
FIN
REQ'D

PERMA RESEARCH & DEVELOPMENT
U.S. RTE 1 NO. ATTLEBORO, MASS
R POISSON 1/21

B-520

[PX 23, p 129]

[PX 23, p 130]



NOTES
1. PITCH DIAMETER OF GEAR MUST BE CONCENTRIC WITH .250-.252 DIA. WITHIN .004 T.I.R.
2. GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

PONTIAC 1960-64
4-SPEED TRANSMISSION

HELICAL GEAR DATA (GENERATING)

PART NUMBER	PERMA NUMBER	NUMBER OF TEETH	IDENTIFICATION		LEAD	PITCH DIA.	CIRCULAR TOOTH THICKNESS (NORMAL)	OVER PIN DIM. REF	PIN DIA
9117625	AK 4PA 20	20	20T	DARK GRAY	9[illegible]6	.7983		[illegible]	0825

NORMAL DIAMETRAL PITCH - - - - 26
NORMAL PRESSURE ANGLE - - - 22°30'
HELIX ANGLE, R.H. - 15°30'34" - - - -
OUTSIDE DIAMETER - - - - .812/.806
BACKLASH WHEN CHECKED WITH MATING GEAR ON 1.006 CENTER - - - - .002-.005 - - -

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK

DO NOT SCALE DIMENSION

SCALE 4X1
MATL SPEC
FAIT
FIN
REQ'D

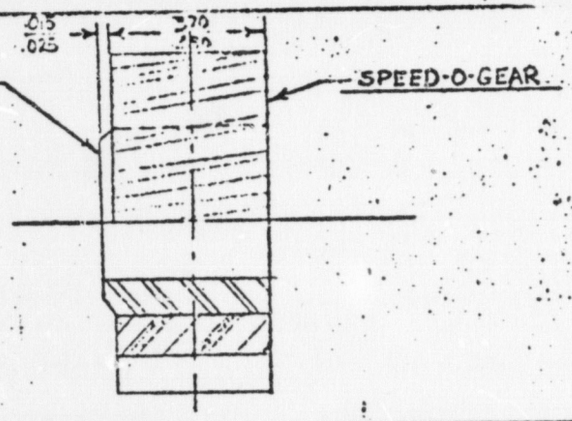
PERMA RESEARCH & DEVELOPMENT
U.S. RTE 1 NO. ATTLEBORO, MASS
TOLERANCES UNLESS OTHERWISE SPECIFIED

B-521

[PX 23, p 130]

B-297

[PX 23, p 131]



NOTES
1. PITCH DIAMETER OF GEAR MUST BE CONCENTRIC WITH .250-.252 DIA WITHIN .004 T.I.R
2. GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

RAMBLER 1960-61-62

HELICAL GEAR DATA (GENERATING)

PART NUMBER	FERMA NUMBER	NUMBER OF TEETH	IDENTIFICATION	LEAD	PITCH DIA	CIRCULAR TOOTH THICKNESS NORMAL	OVER PIN DIM REF
3150562	AKIRA-20	20	GREEN	16.575	.7775	.0540/.0560	

NORMAL DIAMETRAL PITCH	26
NORMAL PRESSURE ANGLE	20°
HELIX ANGLE L.H.	8°23'
OUTSIDE DIAMETER	.856/.851
ROOT DIAMETER	.690/.685
TOOTH DEPTH	.0839

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK

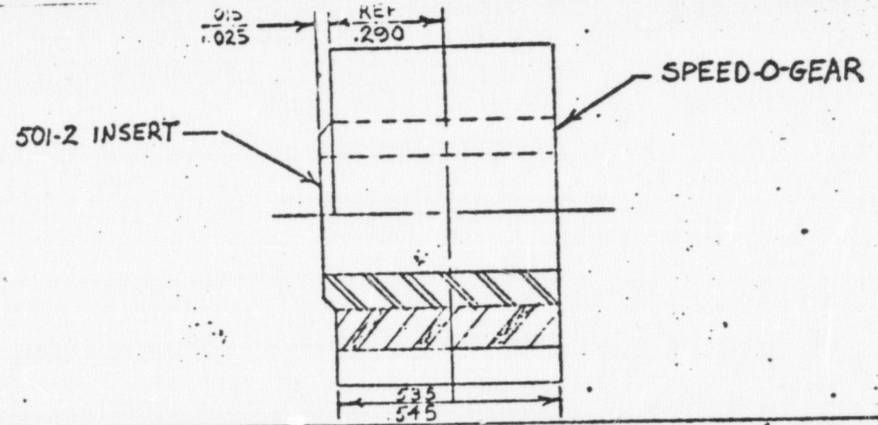
DO NOT SCALE DIMENSIONS

SCALE 2-1
MAT'L
2YTEL

FERMA RESEARCH & DEVELOPMENT CO.
DETROIT AND AUBURN HILLS

B-522

[RX 23, p 131]
E-198



NOTES
1. PITCH DIAMETER OF GEAR MUST BE CONCENTRIC WITH .250-.252 DIA. WITHIN .004 T.I.R.
2 GEAR MUST BE TEMPERED TO PREVENT BRITTLENESS

RAMBLER 1963-64

HELICAL GEAR DATA (GENERATING)

PART NUMBER	PERMA NUMBER	NUMBER OF TEETH	IDENTIFICATION		LEAD	PITCH DIA	CIRCULAR TOOTH THICKNESS (NORMAL)	BASE DIAMETER	WHOLE DEPTH
3167391	AK-1R-21	21	2T	LT BROWN	15.4556	.7583	.0523/.0503	.6999	.0806

NORMAL DIAMETRAL PITCH 28
NORMAL PRESSURE ANGLE 22° 30'
HELIX ANGLE, R.H. 8° 46' 8"
CENTER DISTANCE, OPERATING .1316/.1312
ROOT DIAMETER .660/.655
OUTSIDE DIAMETER .823/.818
BACKLASH .004 TO .011

CON. DETAIL

SCALE 4X1	PERMA RESEARCH & DEVELOPMENT VAN NUYS, CALIF.
MAT'L ZYTEL	DRWN BY R.D.N. 1/21/64
	CHK'D BY
	ENGR'G
	APPROVED

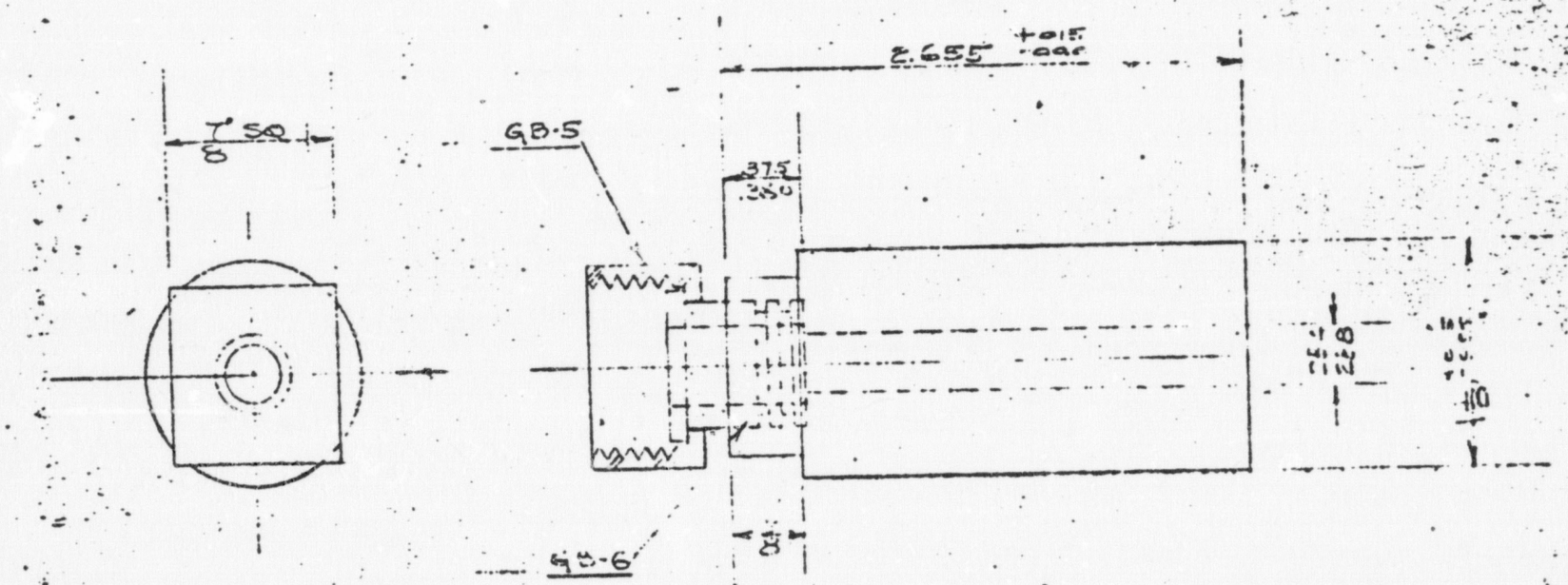
B-523

[PX 23, p 132]

[PX 23, p 132]

E-299

[PX 23, p 133]



NOTE:
OIL ANNEAL TO REMOVE
STRAINS & EMBRITTLEMENT

CONFIDENTIAL

DRWG. TRACING 4771

DATE	LTR.	CHANGE RECORD	BY	CK

DO NOT SCALE DIMENSIONS

SCALE 2/1
MATL STEEL
#1018
FINISH
BEND

SWARE, INC. ATTLEBORO, MASS.

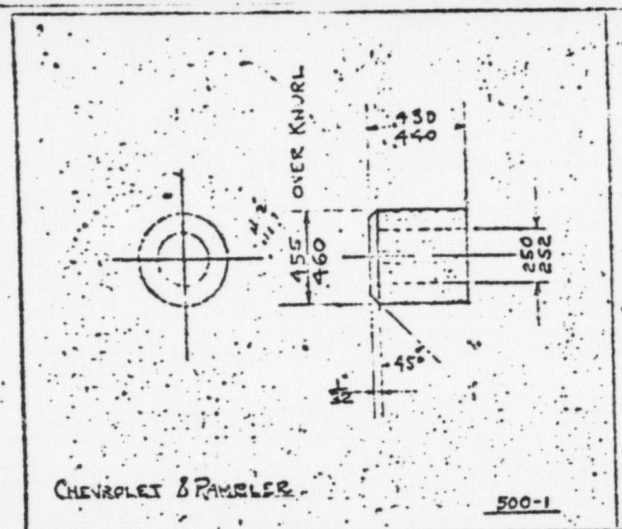
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TRACED
CHECKED
APPROVED

DRAWING NO. 400

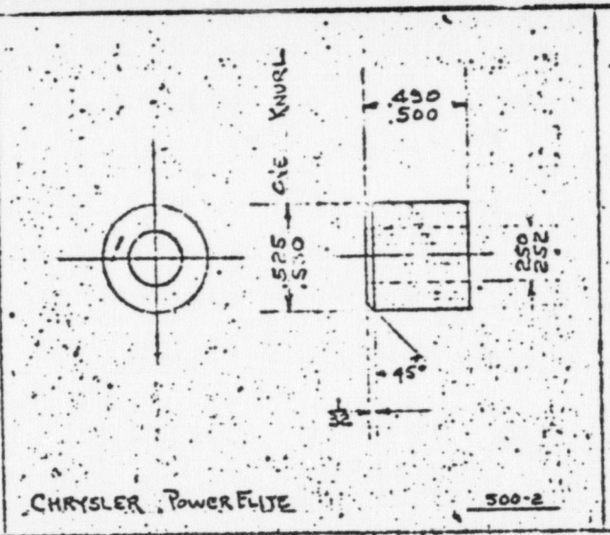
[PX 23, p 133]

E-300

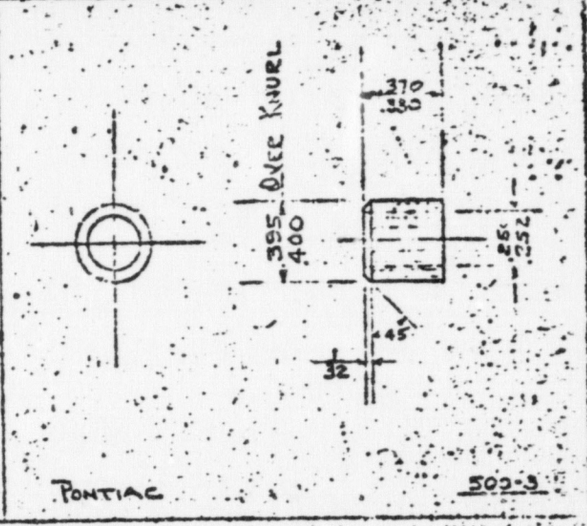
[PX 23, p 134]



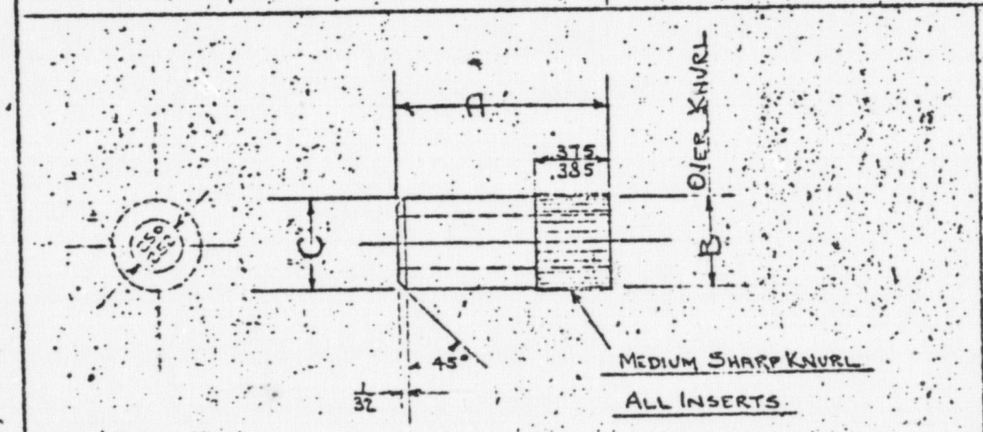
CHEVROLET & RAMBLER 500-1



CHRYSLER PowerFlite 500-2



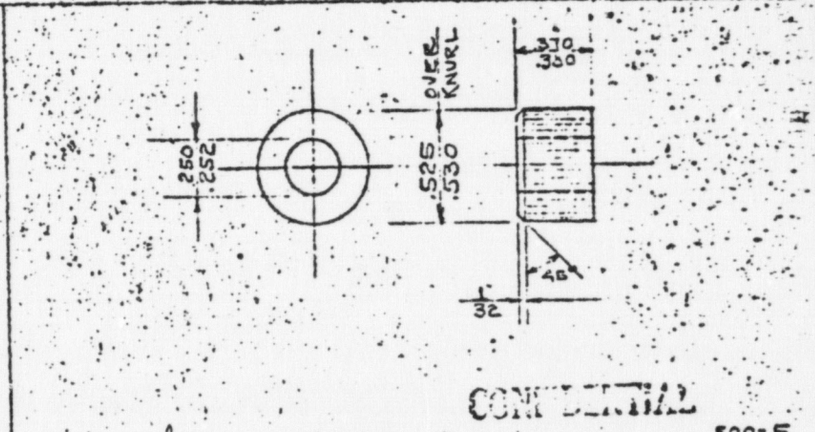
PONTIAC 500-3



MEDIUM SHARP KNURL
ALL INSERTS

500-4

CAR & YEAR	A	B	C
FORD - 1959-1962	.995 1.005	.155 .160	.430 .435
CHRYSLER - 1963	1.120 1.130	.525 .530	.485 .490



ALL BUICKS & CHRYSLER TORQUE FLITE 500-5

CONFIDENTIAL

DATE	LTR	CHANGE RECORD	BY	CK
1-19-62	C	CHRYSLER PT.		
1-26-62	D	ALL 2 SPEED		
2-19-62	E	KNURL SIZE INCREASED		

DO NOT SCALE DIMENSIONS

SCALE 2/1
MATL SAE
DRN
CHK
APPD

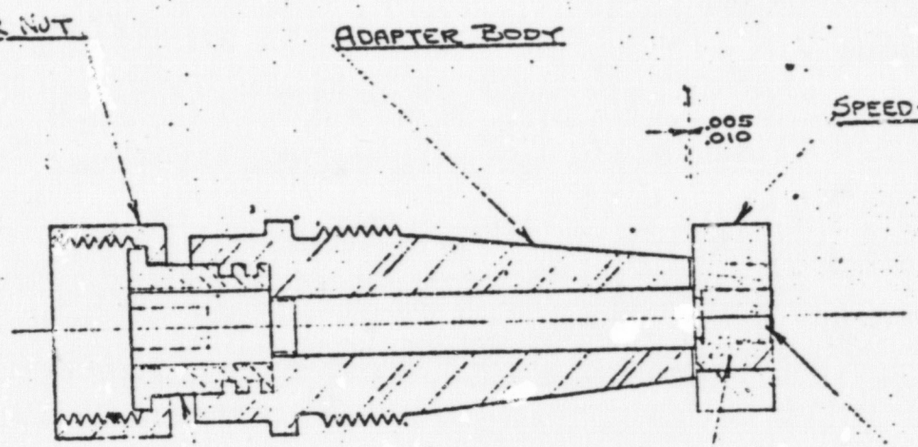
TITLE: SPEEDO GEAR INSERTS

DRAWING NO. 500-C

[PX 23, p 134]

E-301

[PX 23, p 135]



CONFIDENTIAL

						SCALE	
						MATERIAL	
					FOLLOW SHOP INSTRUCTIONS UNLESS SPECIFIED OTHERWISE		
					BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	DATE	
					ALLOWANCE OR TOLERANCE DIMENSIONS	FINISH	
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	HEAT TREAT	

TITLE: Blank Grip S.D. Type [illegible]

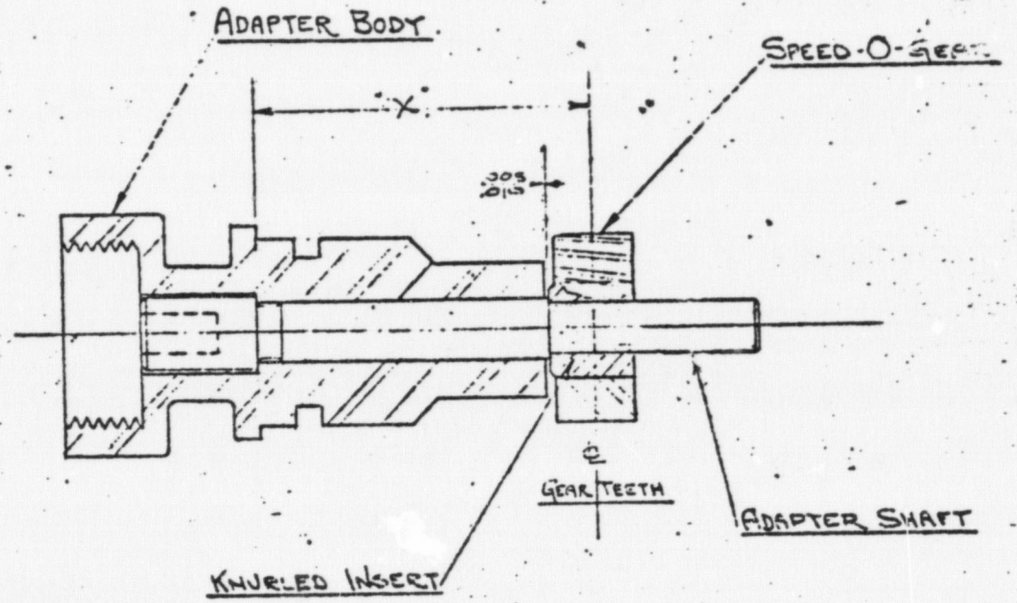
PENN RESEARCH & DEVELOPMENT CO.
[illegible] NO. [illegible] NO. ATTCH. NO. MADE

DRAWING NO. 600

DRAWN BY
CHK BY
CHK BY
APPROVED

[PX 28, p 135]
E-302

[PX 23, p 136]



MAKE OF CAR	"X"	MAKE OF CAR	"X"	MAKE OF CAR	"X"
CHEVROLET 1962-1963	2015	BUICK SPECIAL 1963	1.955	RAMBLER	
CHEVROLET 1961-1962	1.445	BUICK-196_	2.625		
FALCON-COMET 1961-1964	1.560				
VALIANT 1961	1.560	CHRYSLER 626	1.625		
FORD 1961-64	1.560				
PONTIAC 62-63	1.560	PONTIAC 3-SPEED			
LEMANS		PONTIAC 4-SPEED	1.580		

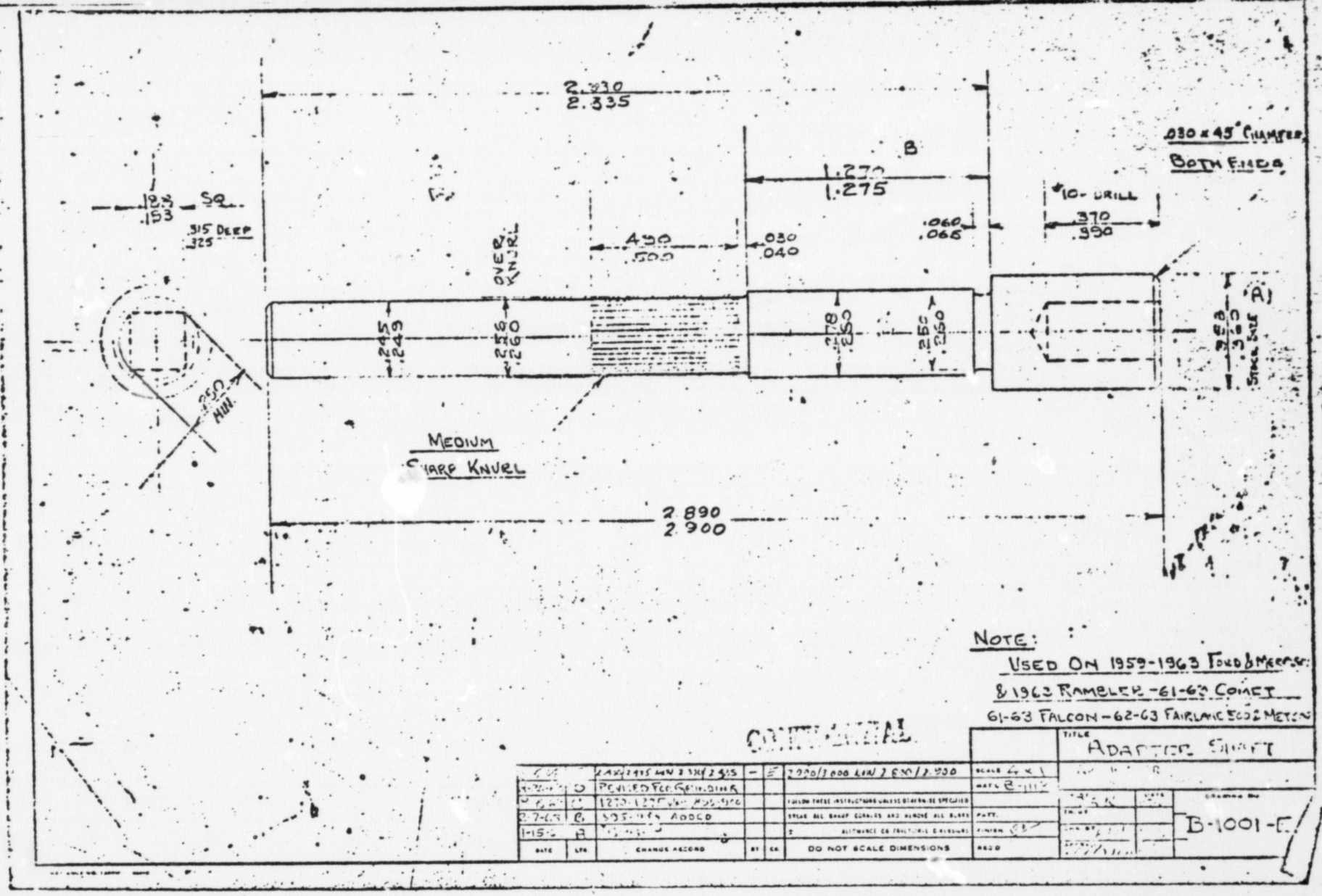
					SCALE 2X	TITLE MOULDED ADAPTER ASSY
					MATL	FELLOW REGULATOR & EQUIPMENT CO.
				FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		UNLESS NOTED NO APPLICABLE MARK
				BREAK ALL SHARP CORNERS AND REMOVE ALL BURR	PATT	DRAWN DATE
				ALLOWABLE ON FRACTIONAL DIMENSIONS	FINISH	CHK'D BY
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	APPROVED

DRAWING NO. 700

[PX 23, p 136]

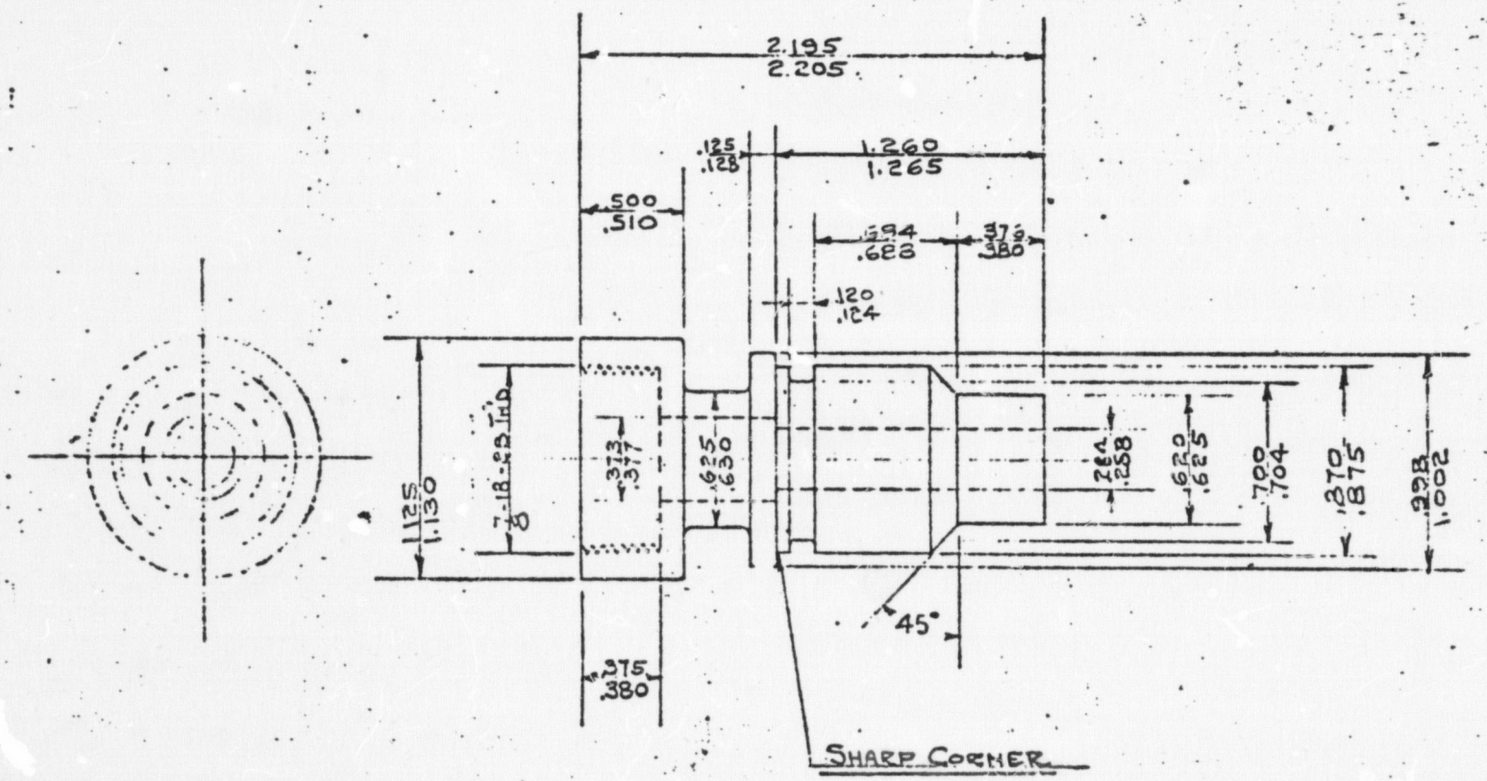
E-303

[PX 23, p 137]



[PX 23, p 137]

[PX 23, p 138]



NOTE:

USED FOR 59-63 GALAXY & FORD WAGONS - 59-63 MERCURY MONTEREY 62-63 FAIRLANE 500 - 62-63 COMET 63 RAMBLER 6 CYL.

CONFIDENTIAL

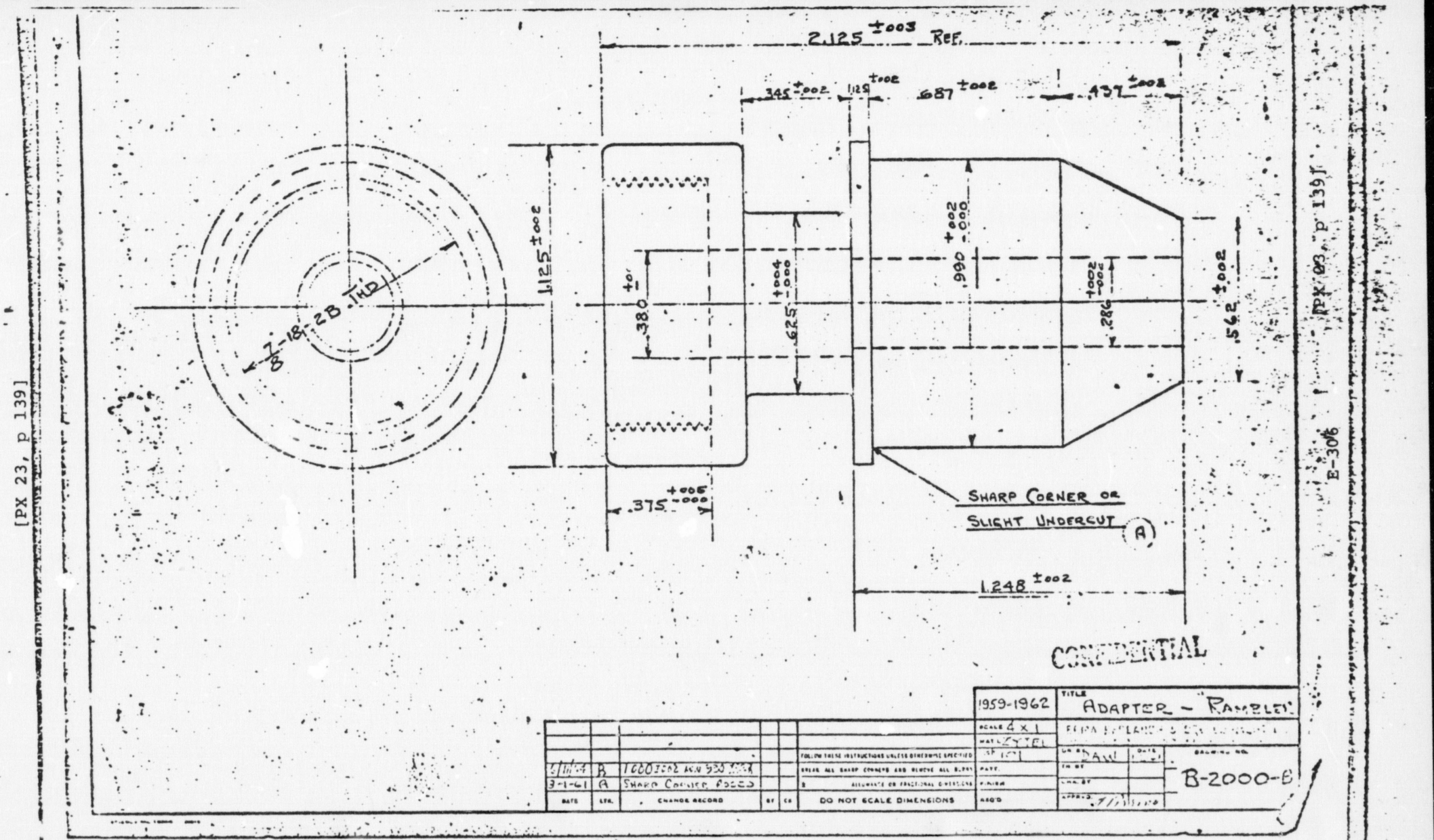
						SCALE 2 X 1	TITLE ADAPTER FOR FORD
						MATL CYTEL 101	PERMA MILLENNIUM & LEVELLOY SUPER LOK U.S. PAT. No. ATTERCOP, MASS.
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DRAWN BY LAH DATE
					BREAK ALL SHARP CORNERS AND REMOVE ALL BURR	FINISH	CHECKED BY
					± ALLOWANCE ON FRACTIONAL DECIMALS		ENGR BY
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MATL	APPROVED

DRAWING NO. B-1100

[PX 23, p 138]

E-305

[PX 23, p 139]



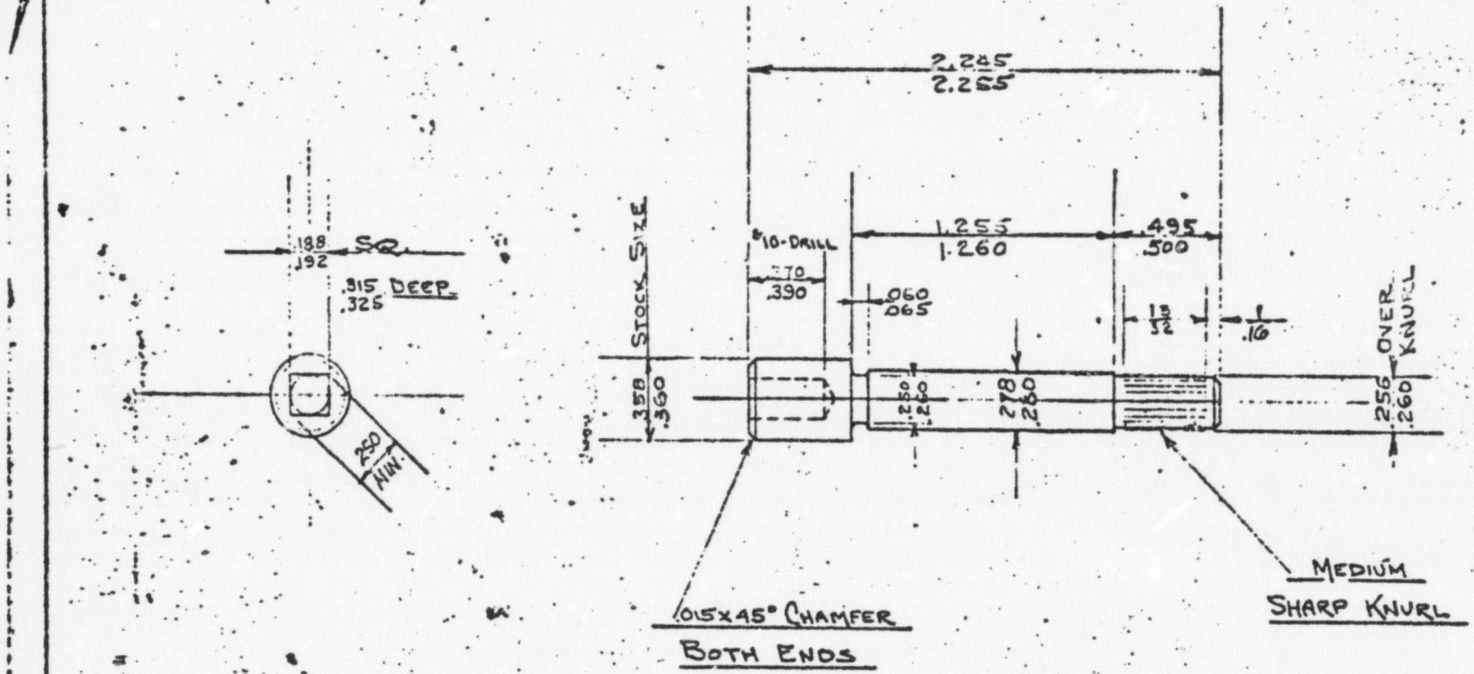
CONFIDENTIAL

						1959-1962	TITLE ADAPTER - RAMBLER
						SCALE 4 X 1	
						MAT'L STEEL	
5/11/64	B	TOOLING NO 530			FOLLOW THESE INSTRUCTIONS UNLESS DIMENSIONS SPECIFIED	ST'D	DRAWN DATE
3-1-61	A	SHARP CORNER ADDED			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PATT.	CH'K'D BY
					ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH	CH'K BY
DATE	LTR.	CHANGE RECORD	BY	CO	DO NOT SCALE DIMENSIONS	ASS'D	APPR'D 7/11/1960

DRAWING NO. B-2000-B

[PX 03, p 139] E-306

[PX 23, p 140]



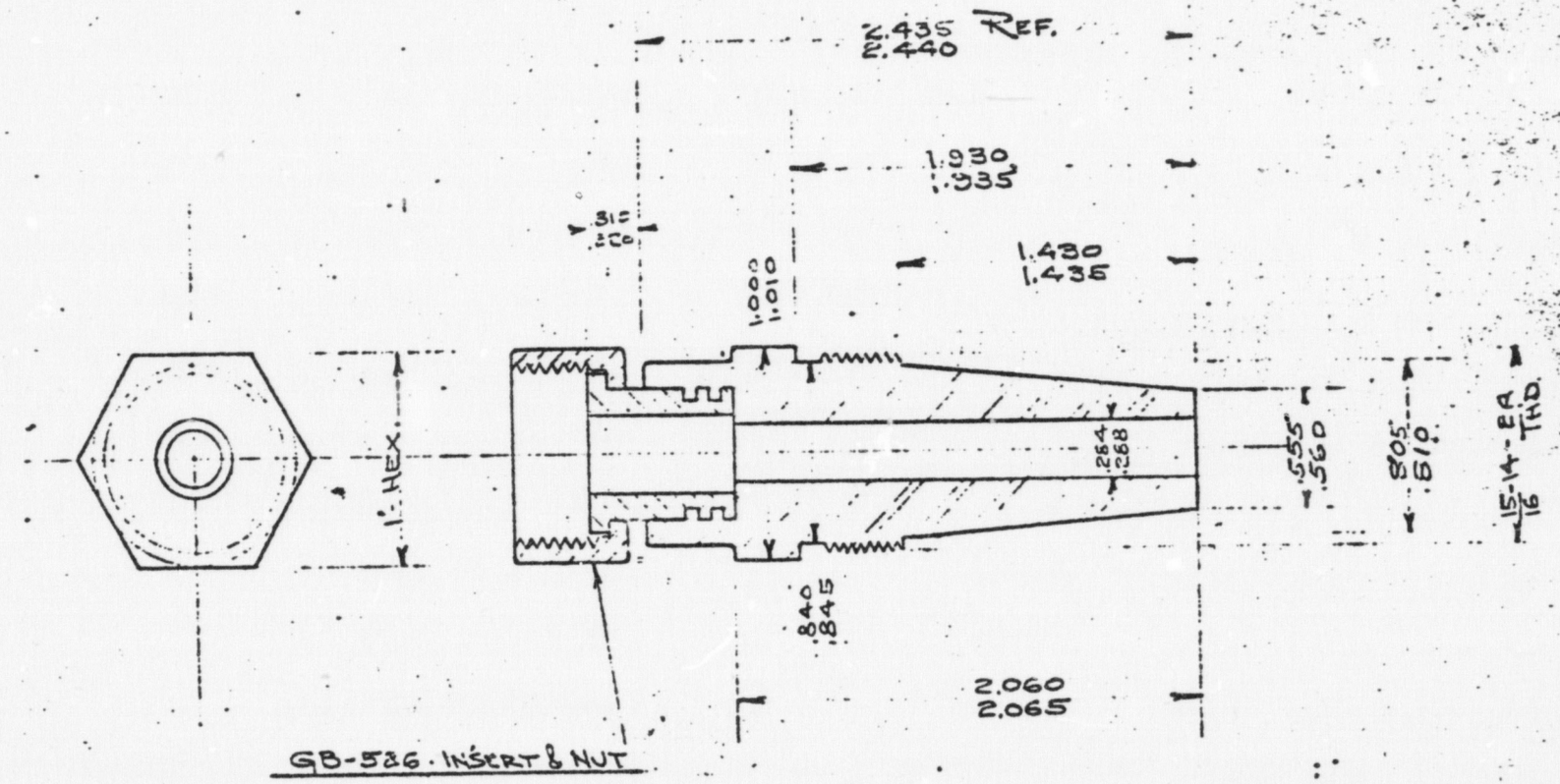
CONFIDENTIAL

						1959-1962	TITLE ADAPTER SHAFT FOR RAMSET
						SCALE 2X1	
						MAT'L B1113	
3-5-63	B	REVISED FOR GRINDING			FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DRAWN DAW
2-7-63	A	THIS WAS 222-			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS		CHK'D
					± ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH	APPR'D
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MISC	DRAWING NO. B-2001-P

[PX 23, p 140]

E-307

[PX 23, p 141]



CONFIDENTIAL

NOTE.

O.D. & THREAD MUST RUN WITHIN .010 T.I.R. MAX. WITH .284-.288 DIA. HOLE

DATE	LTR.	CHANGE RECORD	BY	CK.
10-29-46		REVISED FOR NEW HEAT TREAT		

FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED
BREAK ALL SHARP ... AND REMOVE ALL BURRS
± ALLOWANCE ON FRACTIONAL DIMENSIONS
DO NOT SCALE DIMENSIONS

SCALE 2X1
MATL 2 1/2 FEL
FINISH
ANGLE

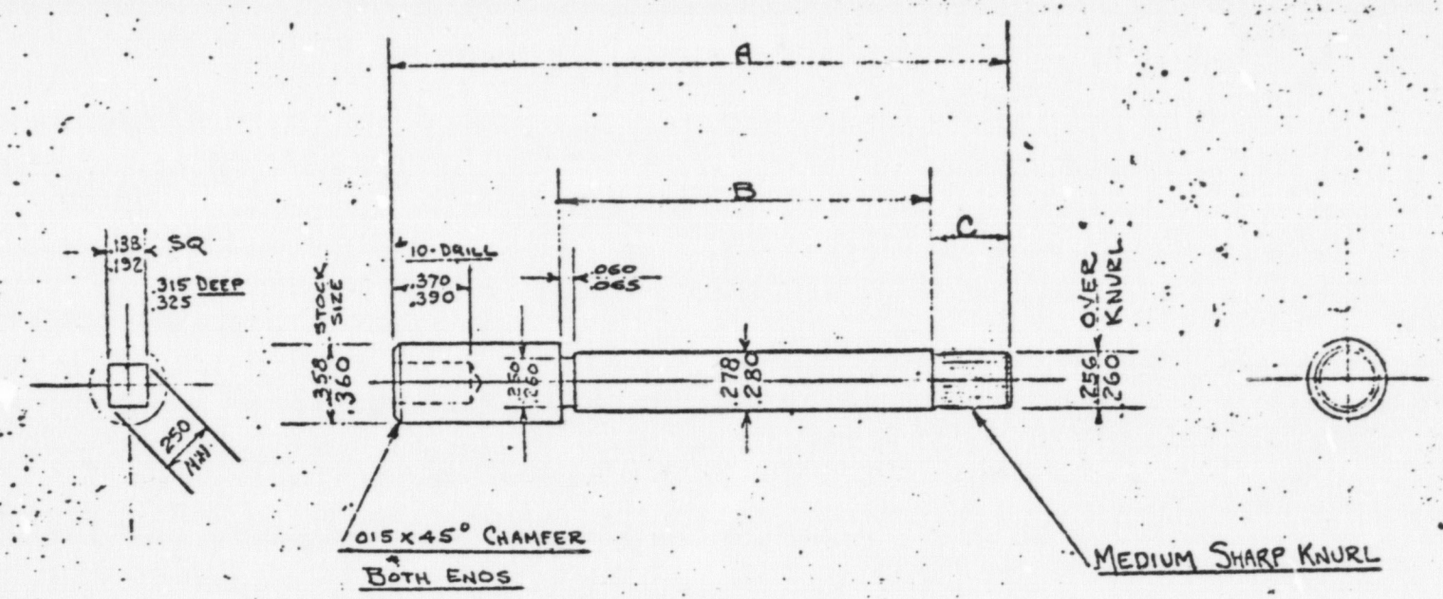
TITLE CHRYSLER-TOWER FLIT...
PERMA RESEARCH & DEVELOPMENT CO. U.S. RTE. 1 NO. ATTLEBORO, MASS.
DRAWN BY DAW
TR BY
CHK BY
APPROVED
DRAWING NO. 3000-A

[PX 23, p 141]
E-308

[PX 23, p 142]

[PX 23, p 142]

B-309

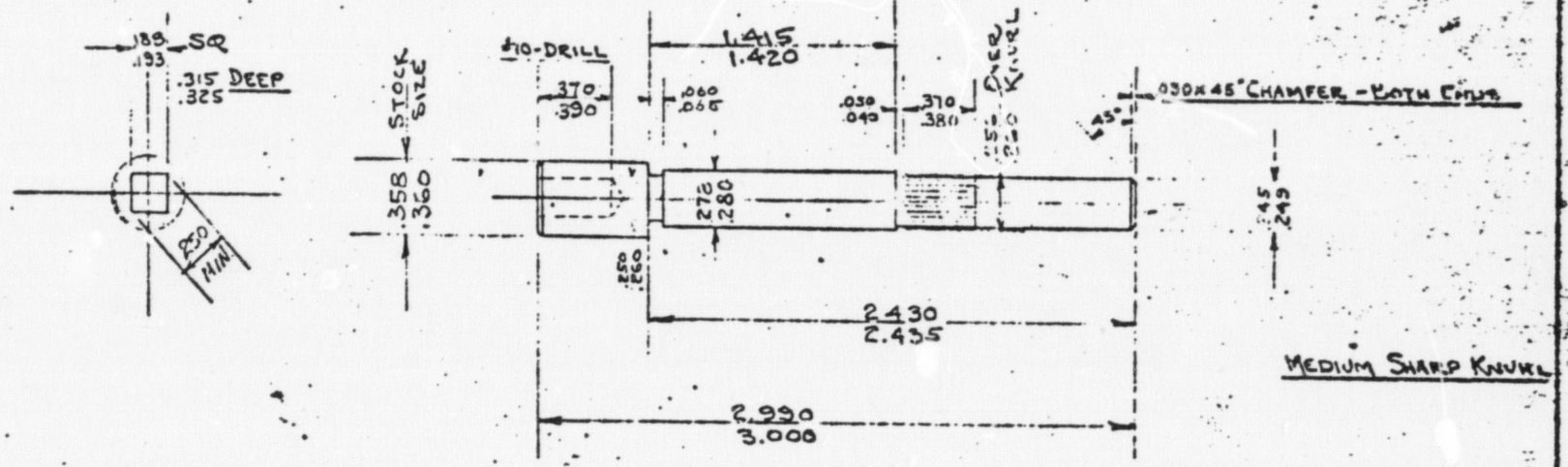


MAKE & YEAR	A	B	C	ADAPTER SHAFT PART NO.
CHRYSLER POWER FLITE	3.250 3.255	2.070 2.075	.495 .500	B-3001-F-1
CHRYSLER TORQUE FLITE	2.685 2.690	1.630 1.635	.370 .375	B-3001-F-2
BUICK - 1959 THRU 1962	2.375 2.380	1.525 1.530	.370 .375	B-3001-F-3
BUICK - 1963	4.795 4.800	2.425 2.430	.370 .375	B-3001-F-4

CONFIDENTIAL

3-30-64	C	REVISED FOR CHANGES	1-27-64	FOR CHRYSLER TORQUE FLITE	SCALE 2 x 1	TITLE ADAPTER SHAFT	
7-7-63	D	SHAFT LENGTHS REVISED			MAT'L 3 1113		
7-1-63	C	SHAFT LENGTHS REVISED		FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DR BY	DATE
1-27-64	B	DIM B CHANGED FOR CHRYSLER		BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	CH'D BY	
9-5-63	A	QUICK FLITE		ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH 63	APPR'D	DRAWING NO. B-3001-F
DATE	LTR	CHANGE RECORD	BY	CK.	DO NOT SCALE DIMENSIONS	REV'D	

[PX 23, p 143]



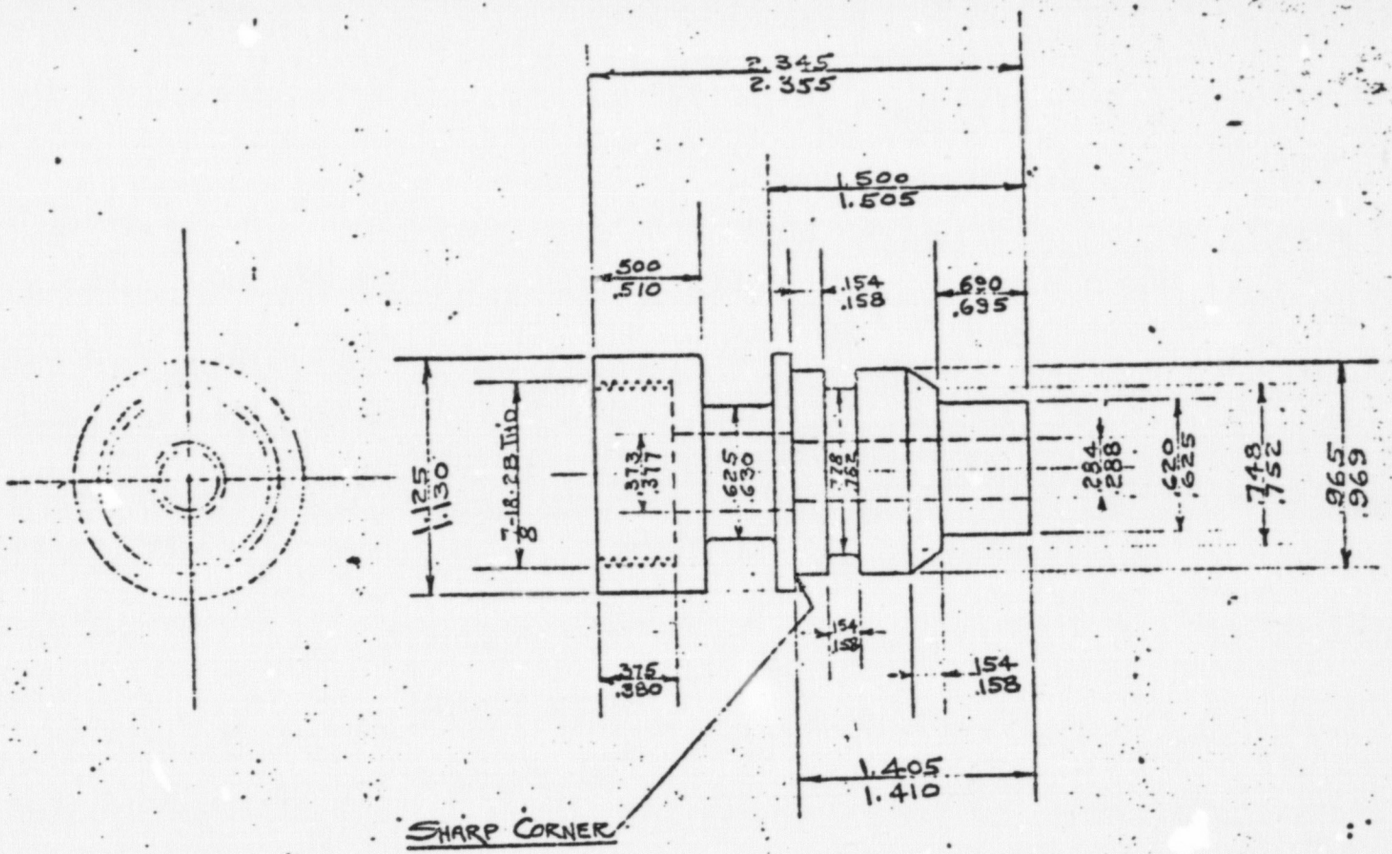
						1962-1963	TITLE ADAPTER SHAFT FOR CHRYSLER
						SCALE 2X	FRAMA RESEARCH & DEVELOPMENT CO. U.S. PAT. & F.C. ANTI-SKID HUBS
2-26-63	C	REVISED TOLERANCES			FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	DWG. B-3113	DRAWN [illegible]
3-15-63	B	REVISED FOR NEW ADAPTER			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PARTS	CHK BY
2-7-63	A	7.5 WAS 7.7			X ± ALLOWABLE ON FRACTIONAL DIMENSIONS	FINISH 63	ENG BY
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MATL'S	APPROVED

DRAWING NO. B3101-C

[PX 23, p 143]

B-310

[PX 23, p 144]



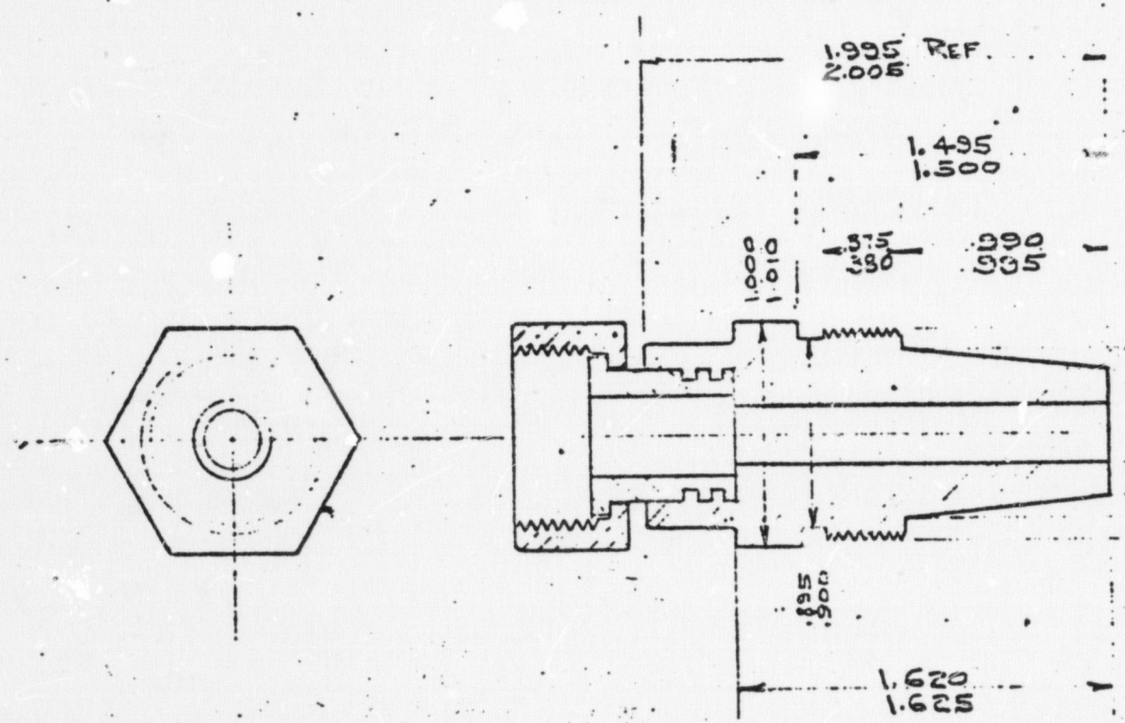
						1962-1963	TITLE ADAPTER FOR CHRYSLER
						SCALE 2/1	GENERAL RESEARCH & DEVELOPMENT CO.
						MATL 7075	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	.101	DRAWN BY [illegible] DATE
					BREAK ALL SHARP CORNERS AND DEBURR ALL BLOCK	FINISH	CHK BY
					± ALLOWANCE ON FRACTIONAL DIMENSIONS		APPD
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	ANSI B	

B-3200

[PX 23, p 144]

E-311

[PX 23, p 145]



NOTE:

O.D. & THREAD MUST RUN
WITHIN .010 T.I.R. MAX.
WITH .254-.288 DIA. HOLE

CONFIDENTIAL

DATE	LTR.	CHANGE RECORD	BY	CK.
10-24-62	B	Revised For New Gap Pin		

DO NOT SCALE DIMENSIONS

FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED
BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS
± ALLOWANCE ON FRACTIONAL & DECIMALS

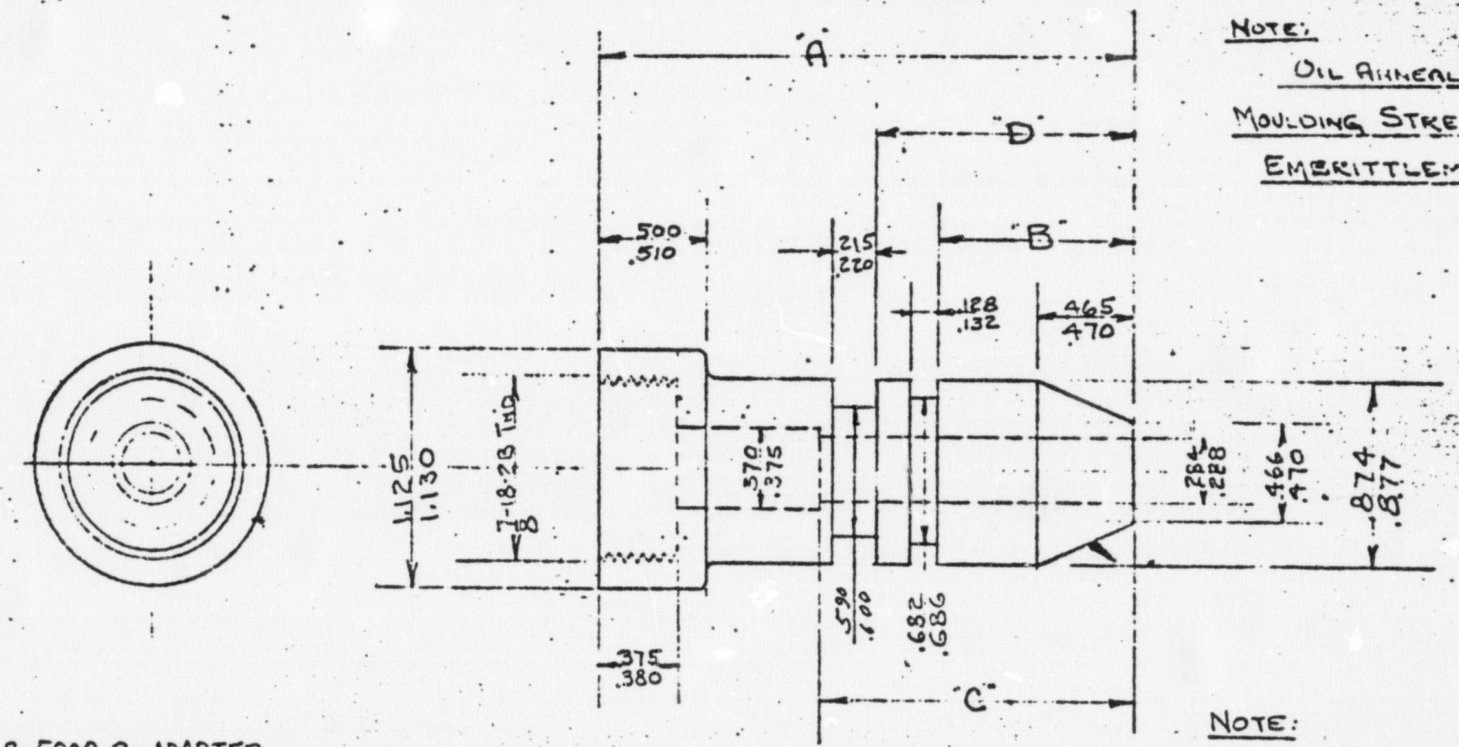
SCALE 2 X 1
MATERIAL
HEAT TREAT
FINISH
BEND

TITLE CHRYSLER TORQUE FLITE - A.D.A.
PERMA RESEARCH & DEVELOPMENT CO.
L P [illegible] NO AFFILIATION WITH
DRAWN
TRACED
CHECKED
APPROVED
DATE
GRAW NO. 4000-A

[PX 23, p 145]

E-312

[PX 23, p 146]



NOTE:
OIL ANNEAL TO REMOVE
MOULDING STRESSES &
EMBRITTLEMENT

NOTE: B-5000-2 ADAPTER
TO BE MACHINED FROM BLANK ADAPTER

NOTE:
O.D. MUST RUN
WITHIN .006 T.I.R. MAX.
WITH .284-.288 DIA. HOLE

YEAR	A	B	C	D
1959-1960	3.103 3.110	1.450 1.455	2.040 2.045	1.790 1.795
1961-1963	3.540 3.552	.915 .920	1.475 1.480	1.225 1.225

B-5000-2
B-5000-1

CONFIDENTIAL

DATE	LTR.	CHANGE RECORD	BY	CK.

DO NOT SCALE DIMENSIONS

SCALE 2 X 1
DATE 2-7-61
FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED
BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS
ALLOWANCE OR FRACTIONAL DECIMAL
FINISH
MATL

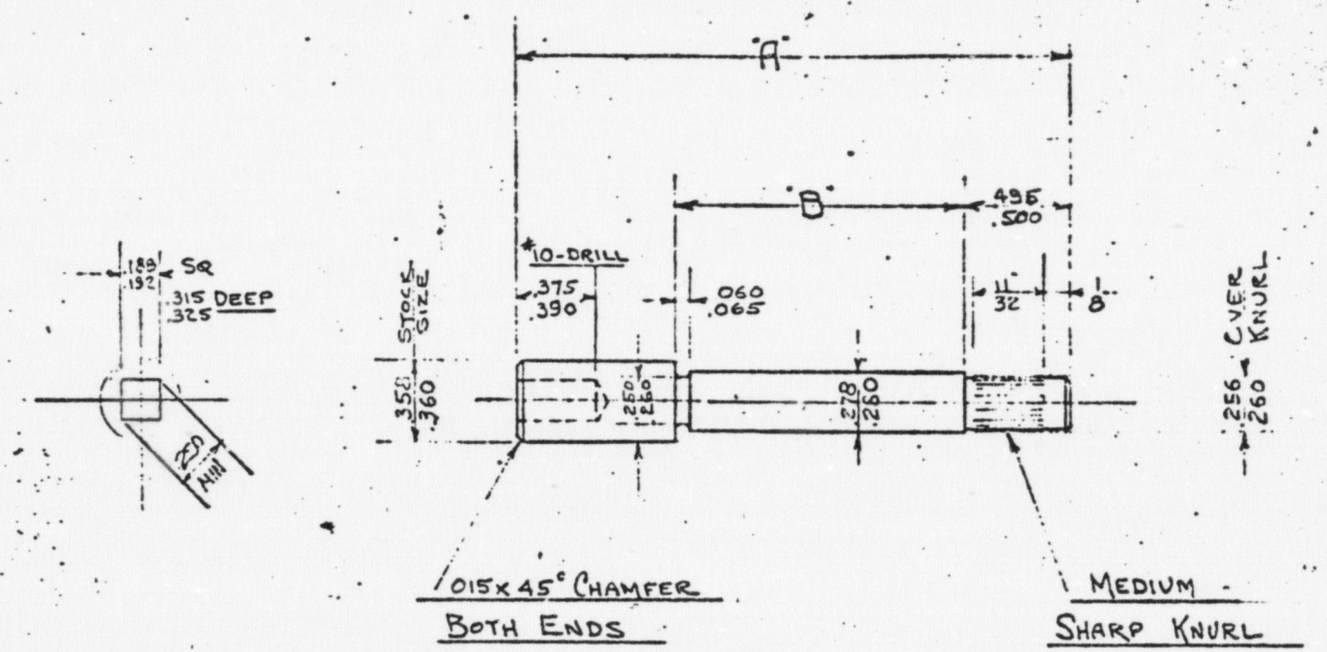
TITLE CHEVROLET ADAPTOR

DRAWING NO. B-5000
1&2

[PX 23, p 146]

E-313

[PX 23, p 147]



CHEVROLET YEAR	A	B	ADAPTER SHAFT NO.
1959-1960	2.670 2.675	1.490 1.490	B5001-C-1
1961-1963	2.670 2.675	1.490 1.490	B5001-C-2
BUICK SPECIAL-GE	2.770 2.775	1.60 1.615	B5001-C-3

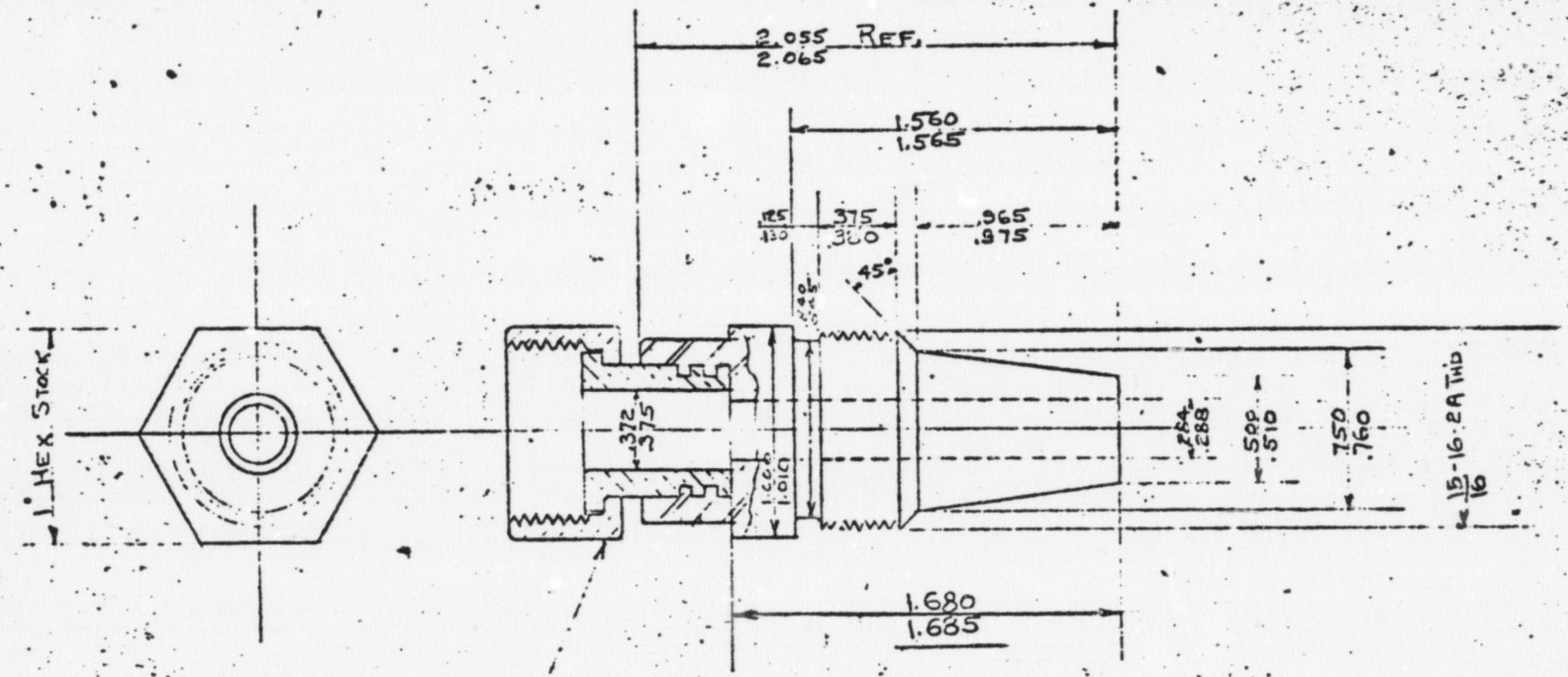
CONFIDENTIAL

						SCALE 2X1	TITLE ADAPTER SHAFT FOR CHEVROLET
						MAT'L B1113	
3-30-61	C	CHANGED FOR CLEARANCE			FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		
1-2-61	B	CHANGED KNURL FOR TOOL			BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PATT.	
10-19-60	A	CHG. P. CHANGED ON CHEV.			± ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH 63	DRAWN [illegible] B-5001-C 1-2-3
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	HEAT TREAT	APPROVED [illegible]

B-314

[PX 23, p 147]

[PX 23, p 148]



NOTE:

O.D. & THREAD MUST RUN WITHIN .010 T.I.R. MAX. WITH .284-.288 DIA. HOLE.

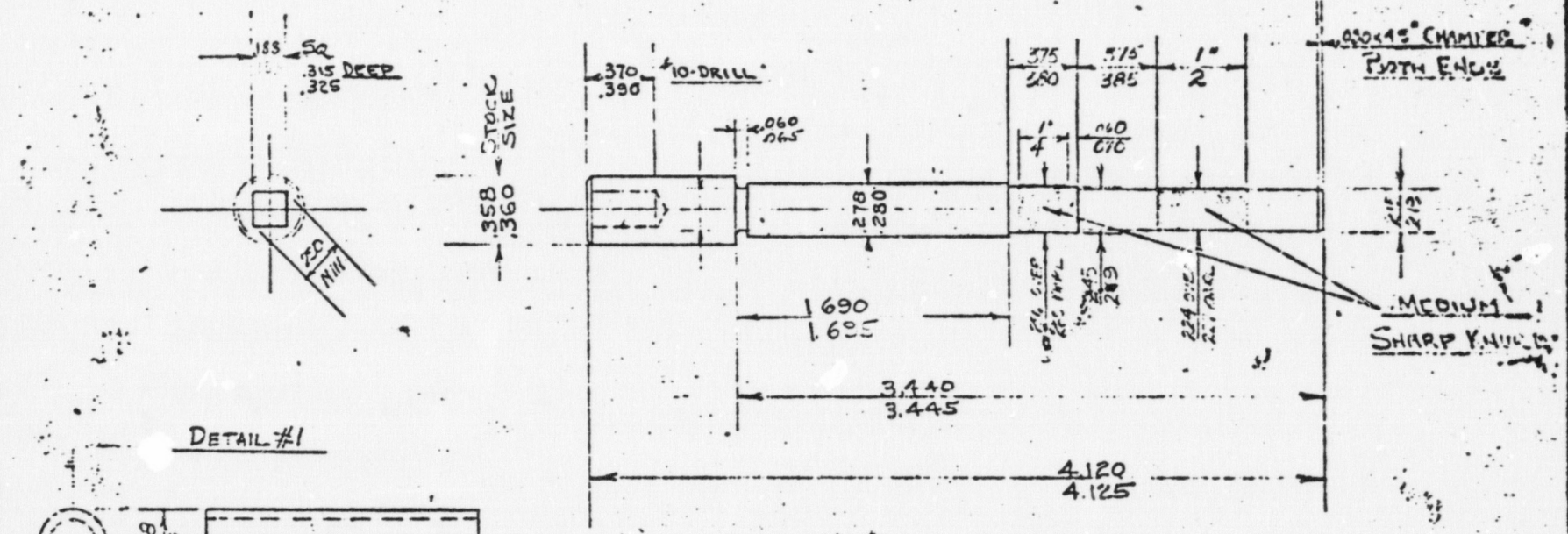
CONFIDENTIAL

						SCALE 2 X 1	TITLE ADAPTER FOR CADILLAC
						MAT'L NYLON	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED	STEEL 101	DRAWING NO B-6000-5
					BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	FINISH	
7-17-63	S	REVISED FOR [illegible] AS BUILT			ALLOWANCE OR FUNCTIONAL DIMENSION		
DATE	LTR.	CHANGE RECORD	BY	CK.	DO NOT SCALE DIMENSIONS	REQ'D	

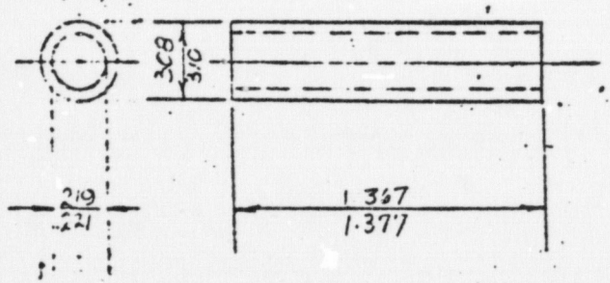
[PX 23, p 148]

E-315

[PX 23, p 149]



DETAIL #1



MATERIAL - BRASS

CONFIDENTIAL

DATE	LTR.	CHANGE RECORD	BY	CK.
5/24/64	C	REDRAWN		
6/8/64	D	REVISED FOR BENDING		
7/27/64	E	REVISED FOR NEW MATERIAL		

DO NOT SCALE DIMENSIONS

FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED
SPECIFY ALL SHARP CORNERS AND REMOVE ALL BURRS
ALLOWANCE ON FRACTIONAL DIMENSIONS

CADILLAC

TITLE: ADAPTER, SHAFT & INSERT

SCALE 2 X 1
DATE 8-11-63
MATL
FINISH
HEAT TREAT
APPROVED

DRAWING NO. B 6100-C

B 6100-C

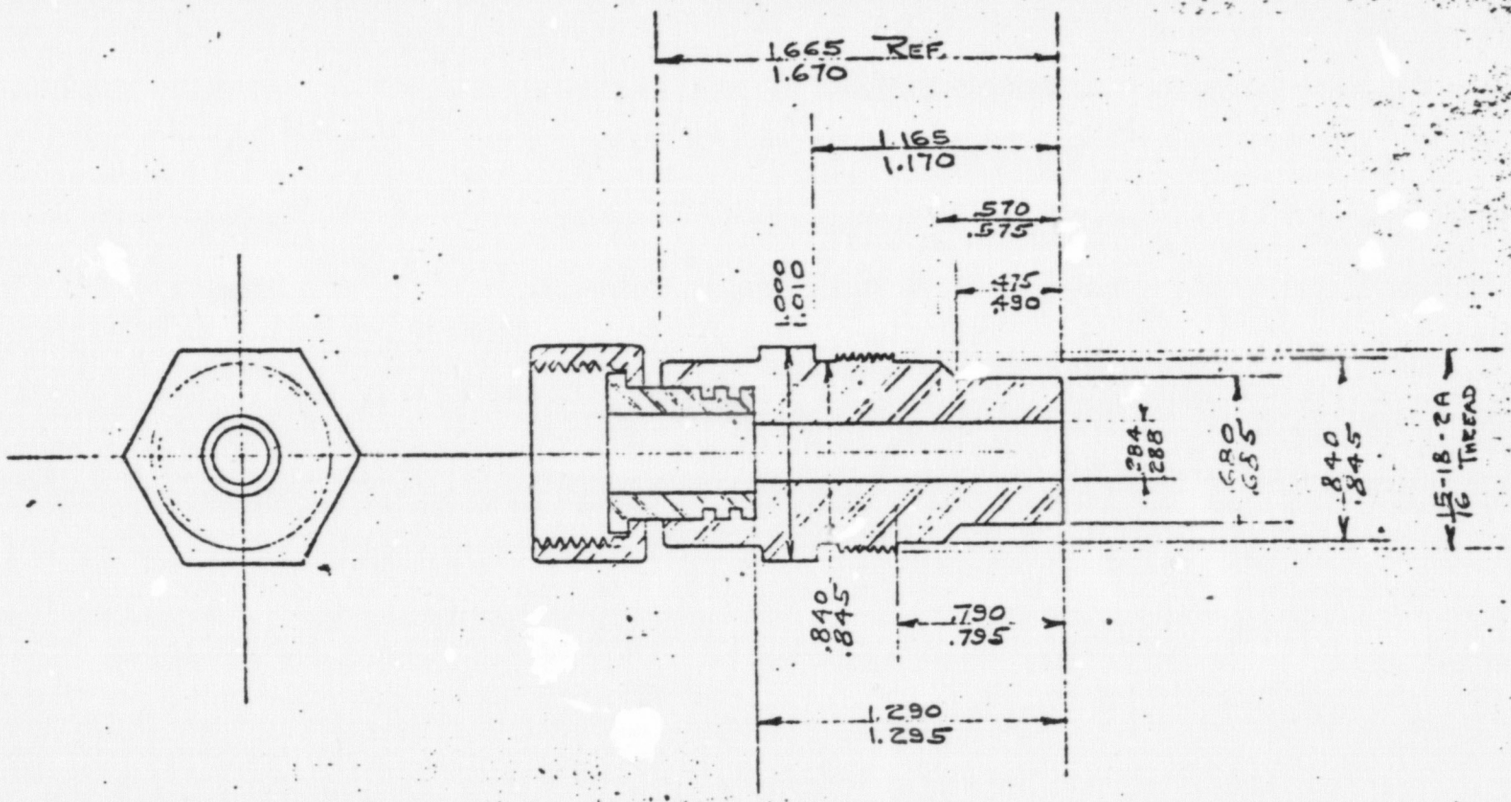
[PX 23, p 149]

E-316

[PX 23, p 150]

[PX 23, p 350]

E-317



NOTE:

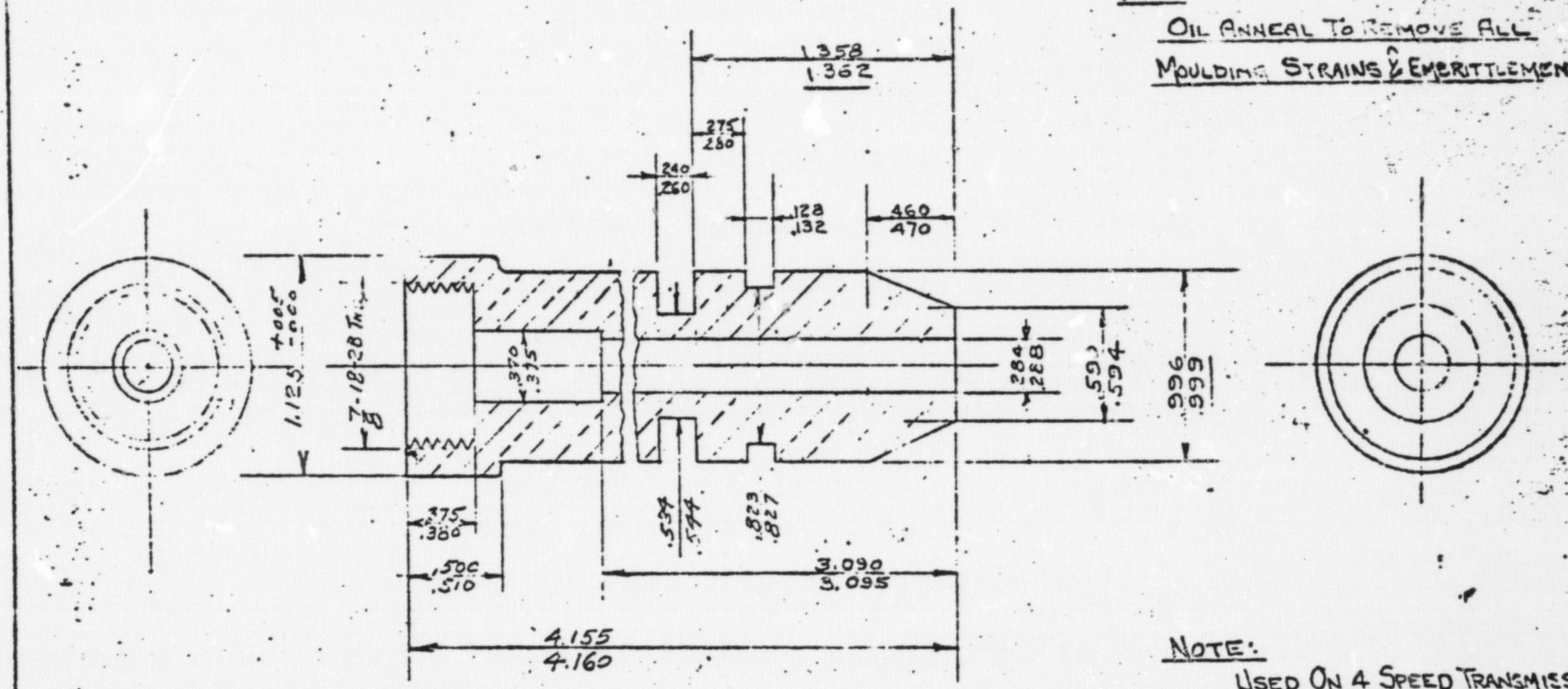
O.D. & THREAD MUST RUN WITHIN .006 T.I.R. MAX. WITH .284-.288 DIA. HOLE

CONFIDENTIAL

								TITLE PONTIAC - 3 SPEED TRANS. ADAPTER	
								PERRIN R. LEARCH & DEVELOPMENT CO.	
							FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		
							BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS		
							± ALLOWANCE ON FRACTIONAL DIMENSIONS		
DATE	LTR.	CHANGE RECORD	BY	CK.	DO NOT SCALE DIMENSIONS				

B-7000-F

[PX 23, p 151]



NOTE:
OIL ANNEAL TO REMOVE ALL
MOULDING STRAINS & EMBRITTLEMENT

NOTE:
USED ON 4 SPEED TRANSMISSION

NOTE:
O.D. MUST RUN WITHIN
.006 T.I.R. MAX. WITH
.284-.288 DIA. HOLE

DATE	LTR	CHANGE RECORD	BY	CK
7/16/51	A	Issued for Manufacture		

DO NOT SCALE DIMENSIONS

ADAPTOR FOR PONTIAC

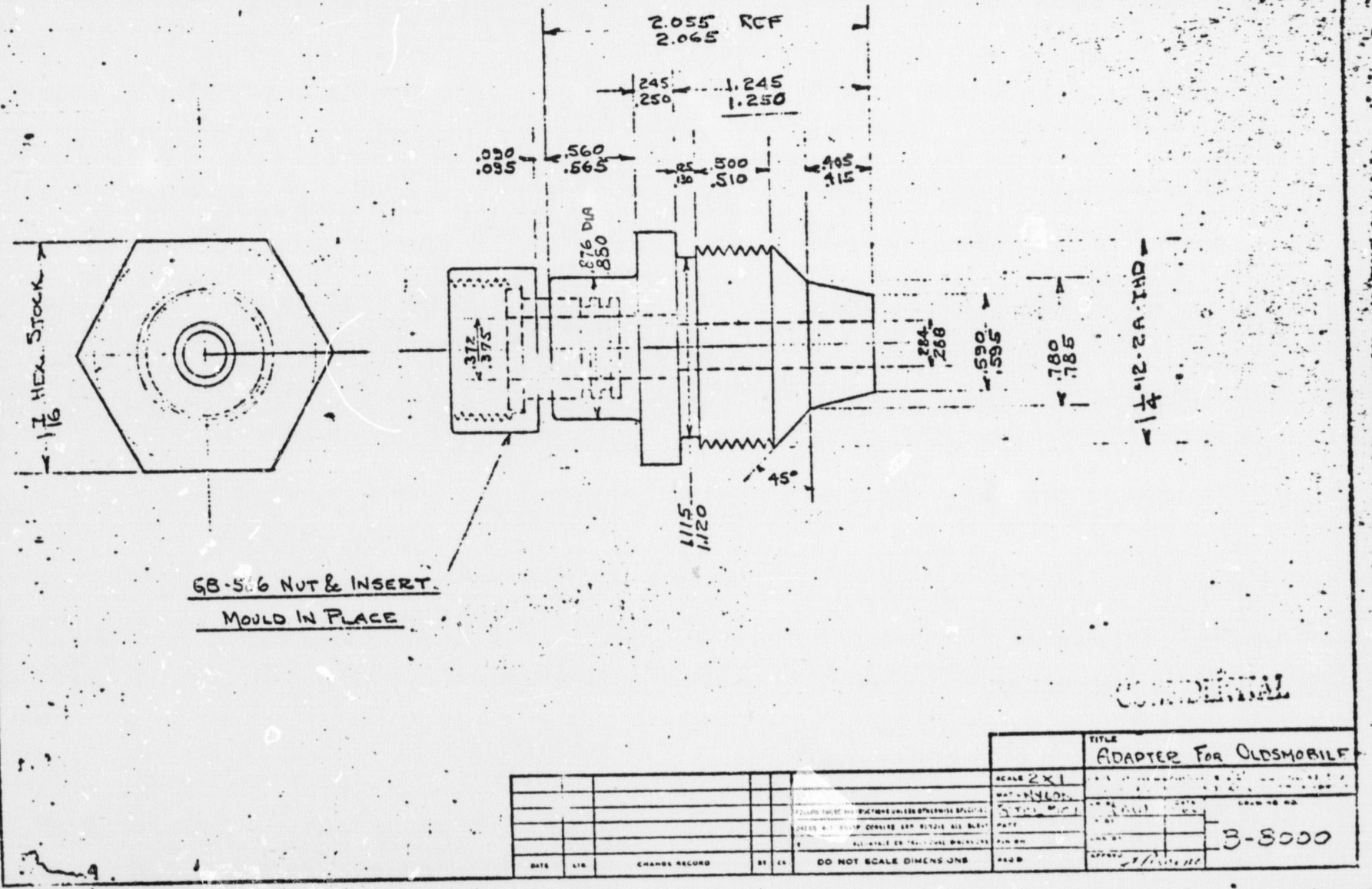
ALPHA RESEARCH & DEVELOPMENT CO.

B-7100-F

[PX 23, p 151]

E-318

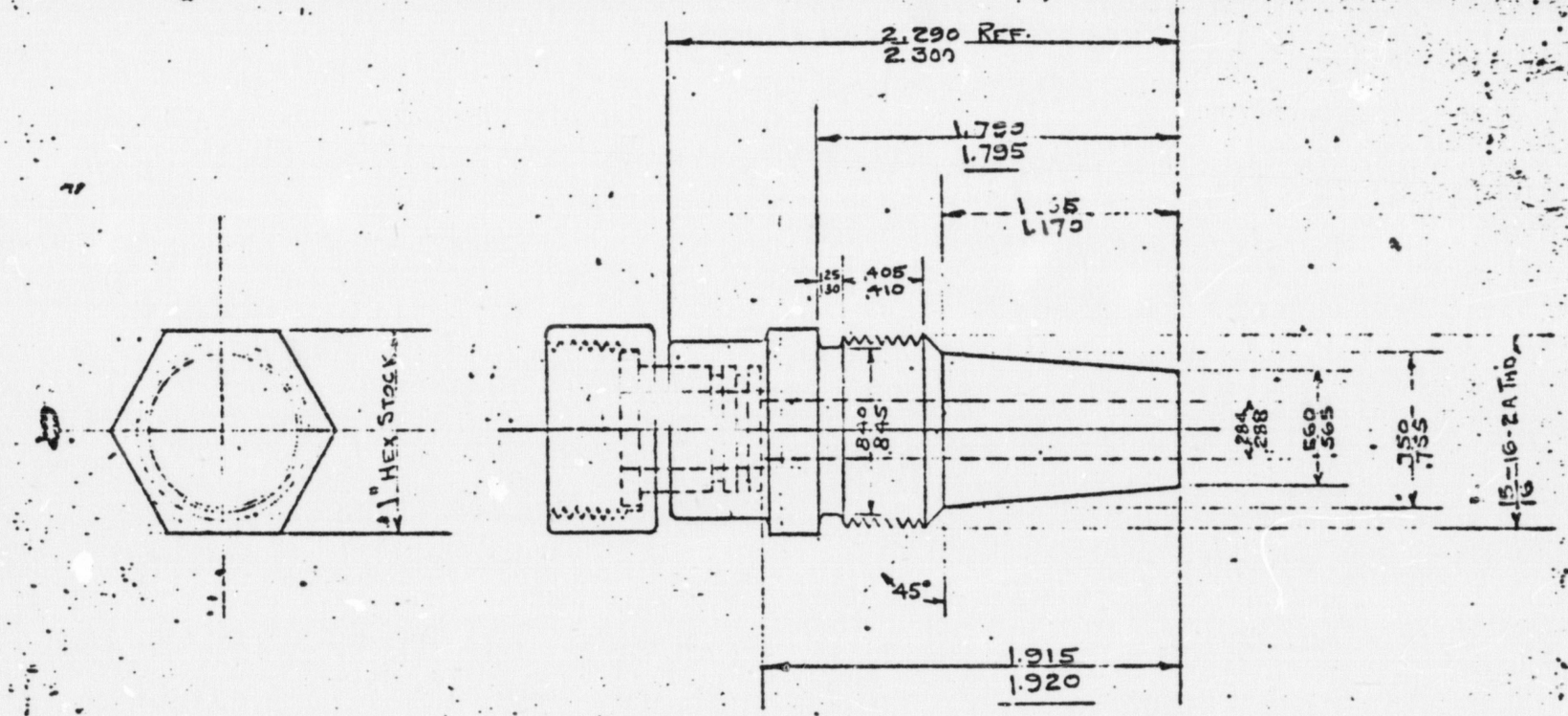
[PX 23, p 152]



[PX 23, p 152]

E-319

[PX 23, p 153]



NOTE:

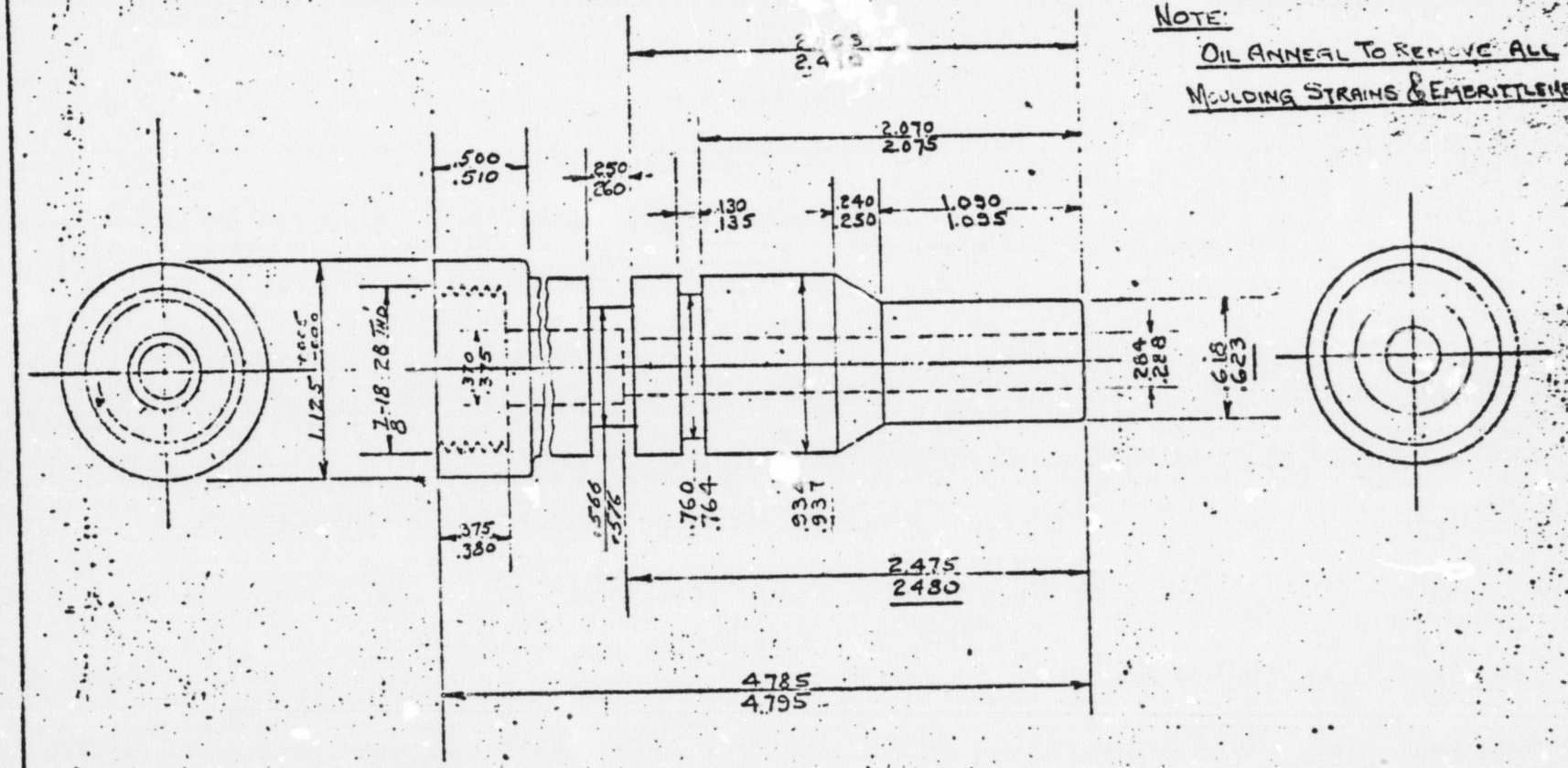
O.D. & THREAD MUST RUN WITHIN .006 T.I.R. MAX. WITH .254-.258 DIA. HOLE

CONFIDENTIAL

							1959-62	TITLE ADAPTER FOR BUICK
							SCALE 2 X 1	
							MATL 4101	
						FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		DRAWING NO B-9000
						BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PART	
5/7/63	H					ALLOWANCE ON FRACTIONAL DIMENSIONS	FINISH	
DATE	LTR	CHANGE RECORD	BY	CK		DO NOT SCALE DIMENSIONS	HEAT TREAT	

[DX 123, p 158]
E-320

[PX 23, p 154]



NOTE:
OIL ANNEAL TO REMOVE ALL
MOULDING STRAINS & EMBRITTLEMENT

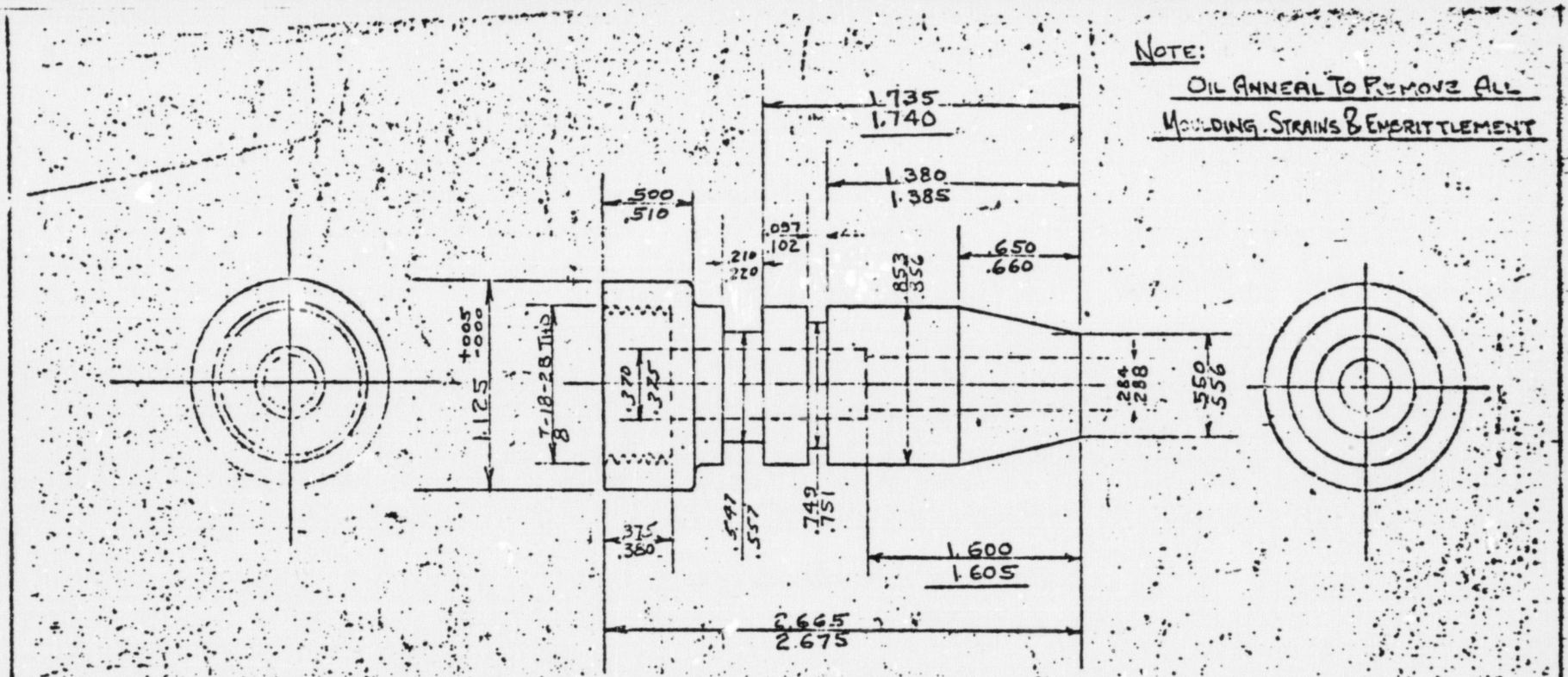
NOTE:
O.D. MUST RUN WITHIN
.006 T.I.R. MAX. WITH
.234-.238 DIA HOLE

CONFIDENTIAL

						SCALE 2X1	TITLE ADAPTER FOR 63 BUICK.
						MATL	
				FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		TITLE	
				BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS		FINISH	
				3 PLACE DECIMALS ± .005 2 PLACE DECIMALS ± .01 ANGLES ± 1°		FINISH	
DATE	LTR.	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	REQD	DRAWN APPROVED DATE DRAWING NO. B-9100-1

[PX 23, p 154]
E-321

[PX 23, P 155]



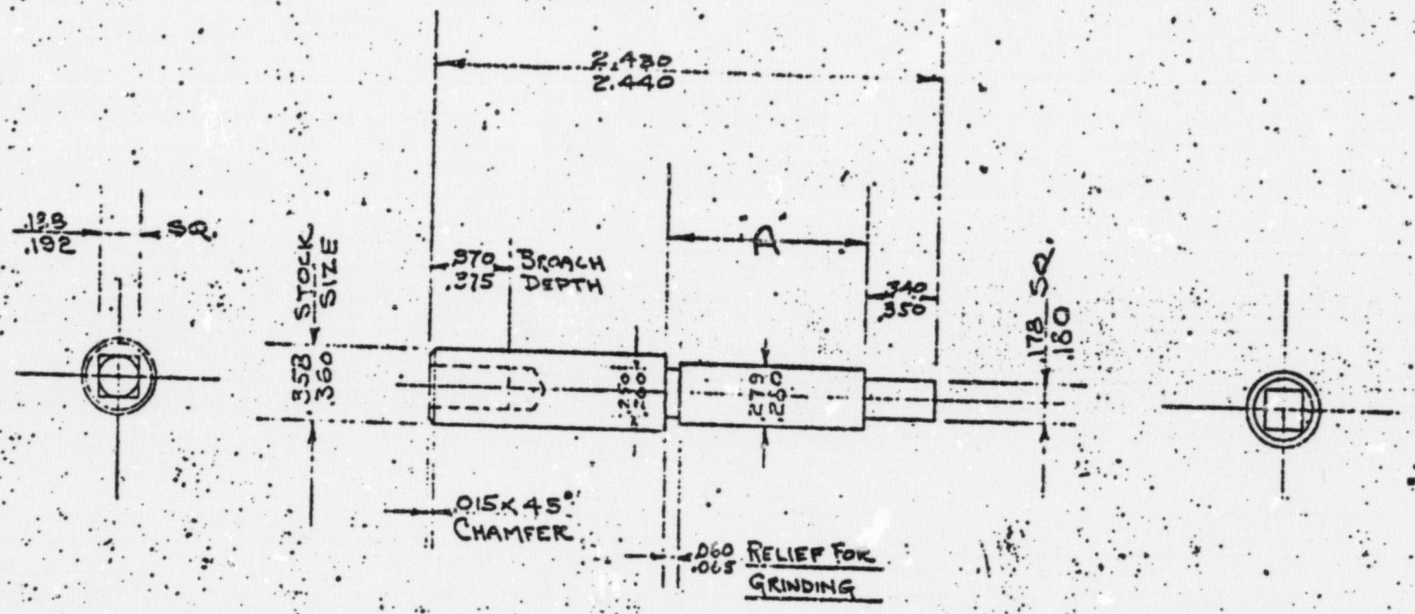
NOTE:
OIL ANNEAL TO REMOVE ALL
MOULDING STRAINS & EMBRITTLEMENT

NOTE:
O.D. MUST RUN WITHIN
.006 T.I.R. MAX WITH
.284-.288 DIA. HOLE

CONFIDENTIAL

						063	TITLE ADAPTER FOR PITCH SERVO
						SCALE 2:1	
						DATE 11-1-1	
					FOLLOW THESE INSTRUCTIONS UNLESS OTHERWISE SPECIFIED		
					BREAK ALL SHARP CORNERS AND REMOVE ALL BURRS	PART	
					X DECIMALS ± FRACTIONAL ± ANGULAR ±	FIN ON	B-9200-1
DATE	LTR	CHANGE RECORD	BY	CK	DO NOT SCALE DIMENSIONS	MATL	

[PX 23, P 155]
E-322



	A
MAKE FOR OLD BLANK ADAPTER	.960 .955
MAKE FOR NEW BLANK ADAPTER	1.398 1.403

CONFIDENTIAL

DO NOT SCALE DIMENSIONS

SCALE 2X1

TITLE SPECIAL ADAPTER CUTTER

PERMA RESEARCH & DEVELOPMENT CO.

AC-1

U.S. EXHIBIT
S.D. DIST. COURT
N.Y.
23
E-322A
A. Daniel Fusaro
[PX 23, p 156]
[PX 23, p 156]

DEF. Ex. 100

PLAINTIFF'S EXHIBIT 24

Mechanical Engineering Division
May 22, 1964

ANTI-SKID DEVICE

Test Specifications for Pressure Switch Equipment

P.O. S5287 — Job # C-701.150

EQUIPMENT:

(1) Fixture to hold switch, quick connect to hydraulic system and electrical system. Airoyal Proposal 5023

(2) Hydraulic power pack capable of developing 3 to 4,000 PSI, with small volume.

(3) Necessary valves and gages to apply, regulate and indicate hydraulic pressure to switch.

(4) Electrical equipment to signal opening and closing of pressure switch contacts.

(5) Hydraulic fluid is to be UCON BF 110861 (UNION Carbide)

PROCEDURE:

(1) Load switch into fixture and clamp (connect hydraulic and electrical lines).

(2) Rise hydraulic pressure to switch until 1,000 PSI is reached.

(3) Allow pressure to fall slowly. Set pressure switch so that electrical contacts open at

PROCEDURE: (Cont'd)

85 to 100 PSI ± 5 PSI falling pressure. (Check for repeatability of final switch setting at least 3 times).

(4) With switch set as described above, rise hydraulic pressure slowly. Electrical contacts should make at 140 PSI ± 10 PSI.

(5) Rise hydraulic pressure to 2,000 PSI and check effect on switch setting. (Contacts should open and close at same setting).

(6) Rise hydraulic pressure to 3,000 PSI and check effect on switch setting. (Contacts should open and close at same setting).

AS per R. Rainelli (Permat)

J. Wagner

Mechanical Engineering Division
May 22, 1964

PRESSURE SWITCH:

The following items were discussed as areas which should be checked but are not included in the test procedure at this time.

(1) Check for possible leaking of fluid past "O" ring. DIMENSIONS & FINISH

(2) Check on type of material and finish on piston #39.

(3) Check for straightness of electrical posts #42 and #43.

(4) Perform life test.

PLAINTIFF'S EXHIBIT 25A

SINGER
561 FIRST STREET ELIZABETH, N.J.

MEMORANDUM

TO: Mr. R. A. Kloby - Dept. 166

FROM: Mr. R. R. Patten - Sales Manager

DATE: May 27, 1964

REFER TO:

SUBJECT: PERMA RESEARCH & DEVELOPMENT COMPANY
MARKET AND FORECAST

PLAINTIFF EXHIBIT
U. S. DIST. COURT
E. D. OF N. Y.
25
PX 25A
11-8-73
clean copy as 25A

A meeting was held at Singer-Elizabeth on Thursday, May 21, 1964 between Mr. Sam Karp, Vice President of Marketing for the subject company, and myself to discuss the firm orders held by Perma and their forecast and marketing plans.

Perma presently has four distributors. Of these four, only one is an exclusive - the one for New England. There are two non-exclusive distributors; one covering Iowa, with an option on Texas, and one in the Minneapolis-St.Paul area. The fourth distributor is in the Dayton, Ohio area and has no formal agreement with Perma.

The firm business applicable to the above groups, plus commitments to Canada prior to production in Canada, are as follows:

Area	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Total
New England	500	500	2500	2500	2500	2500	3500	3500	18,000
Iowa	-	200	200	200	200	300	300	300	1,700
Minn.-St.Paul	-	200	200	200	200	300	300	300	1,700
Dayton*	200	200	200	200	200	200	200	200	1,600
Canada	750	750	750	750	750	-	-	-	3,750
TOTAL	1450	1850	3850	3850	3850	3300	4300	4300	26,750

* This forecast not by agreement but based on area's past performance.

The above represent the total requirements of Perma as of this date through the end of 1964. Singer will, obviously lose the May requirements and probably the June requirements. Some of this may be picked up due to lack of shipments at the present time because of the current production problems.

(last paragraph transcribed on page E-326A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last paragraph thereof reads as follows:

> "The above represent the total requirements of Perma as of this date through the end of 1964. Singer will, obviously, lose the May requirements and probably the (?) requirements. Some of this may be picked up due to lack of (?) at the present time because of the current production problems."

Any actual forecast for additional business does not exist. In our discussion, two significant prospects were disclosed which fall into a forecast-type situation and have at least a 50-50 chance of maturing.

The first of these is a potential with Phillips Petroleum being handled by the Iowa representative. Phillips has an accessory distribution system through jobbers to supply their company-owned stations, as well as private stations, with tires, batteries, etc. They have shown an interest in the Perma Anti-Skid unit and are evaluating it for distribution by their stations. A portion of this evaluation was a market study to determine the usage in their owned stations. The accessory distribution system is handled through 2,104 jobbers. Phillips has a total of 23,362 dealers. They grade their dealers by Class A, B, and below B their classification is by dollar volume. The market survey showed a monthly usage of Perma Anti-Skid of 19,181 units broken down as follows:

Class	No. of Stations	Units/Mo.	Total
A	2,500	2	5,000
B	7,500	1	7,500
Below B	13,362	1/2	6,681
		TOTAL	19,181

This represents a very substantial piece of business and would use over 90% of the original estimate of 250,000 units per year. Of interest, was the pricing structure being discussed between Perma, its distributor and Phillips:

Perma -	Distributor price to Phillips	$ 59.85
	Phillips to Jobber	79.52
	Jobber to Dealer	93.56
	Dealer to Consumer	129.95 plus 10.00 installation

Phillips would be the buying agent and would permit the customer to purchase on his Phillips credit card with a six-month payoff.

The second prospect, which is a possible forecast, is with the United States Post Office Department. The Post Office Department has 4200 trucks purchased on contract from Studebaker that they cannot use. The weight distribution in the truck is such that on stopping from an low a speed as

(last two paragraphs transcribed on page E-327A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last two paragraphs thereof read as follows:

> "Phillips would be the buying agent and would permit the customer to purchase on his Phillips credit card with a six-month payoff.
>
> "The second prospect, which is a possible forecast, is with the United States Post Office Department. The Post Office Department has 4000 trucks purchased on contract from Studebaker that they cannot use. The weight distribution in the truck is such that on stopping from as low a speed as . . . "

5 mph, the rear wheels leave the ground 4 to 6 inches. The various states refuse to permit the vehicles to be driven and the Post Office drivers refuse to use them. Four trucks were equipped with Anti-Skid units and the problem is resolved. The Post Office tells Perma they have no money budgeted for the units, but are requesting the money for fiscal year '65. If the money is budgeted, it is expected that the entire 4200 units will be shipped by December, 1964.

In addition to the Post Office prospect, Perma is working with the Army Mobility Command, General Services Group, on a Jeep brake problem. The Jeep tends to lock up. At present, the General Services Group of the Army Mobility Command has four units on Jeeps for evaluation. To date, no potential commitment or report has been received. In total, some 175,000 Jeeps are involved.

Perma is presently negotiating with several of the automotive companies. The status of each is as follows:

1. FORD MOTORS

Perma purchased a 1964 Ford, equipped it with the anti-skid device and shipped it to the Mercury Division of Ford for evaluation. Tests are presently being conducted. Ford is interested in the Perma Anti-Skid as standard equipment on the 1967 Mercury Marauder. The '67 Marauder is to have discs on the front and drums on the rear. Ford, if it accepts Perma's unit, will offer it as an option on all products. It is expected the results of the evaluation are 4 to 6 months away.

2. CADILLAC DIVISION - GENERAL MOTORS

Presently have 1964 Cadillacs equipped with anti-skid unit. Has been tested by Cadillac Division. This car has been turned over to Delco Marion Division which has control over GMC braking systems.

3. CHEVROLET TRUCK DIVISION - GENERAL MOTORS

Have equipped small truck with unit and tested. Developed a problem when truck was loaded. Perma making corrections and Chevrolet Truck will re-evaluate.

4. GENERAL MOTORS TECH. CENTER

Carrying on additional tests and evaluations.

E. AMERICAN MOTORS

Perma has furnished a unit which is being installed for tests and evaluation.

So far, the automotive manufacturers have neither accepted or rejected the unit. Perma is optimistic and feel they are further ahead with Ford. Perma, so far, has struck out with Chrysler.

The Perma Anti-Skid unit is available for all American-made cars from 1960 on, except for Oldsmobile. The Olds cannot use the unit because the speedometer pick-up is on the left front wheel, rather than from the transmission.

Perma's distributor agreements exclude government, national fleet, national chain stores, and OEM business. Perma plans to expand its sales organization to cover this type business. The timing for such expansion will be a function of getting quality product in sufficient quantity to satisfy their present commitments.

Mr. Karp claims to have an additional 12 to 14 prospective distributors of which he feels he can close approximately 50 per cent.

We questioned Mr. Karp on how soon he would require 5,000 units per week. He claims that within 4 months of receiving a quality product he can expand his sales to a volume of at least 5,000 units per week.

R. R. PATTEN

RRP/dn

cc: Mr. C. S. Morris
Mr. C. W. Brooker

PLAINTIFF'S EXHIBIT 27

DATE: March 7-64

AGREEMENT

AGREEMENT MADE by and between PERMA RESEARCH AND DEVELOPMENT COMPANY, a Delaware Corporation, with its principal place of business in North Attleboro, Commonwealth of Massachusetts, hereinafter referred to as "PERMA" and JANDRO EQUIPMENT CO. 705 WEST 77½ ST. MINNEAPOLIS 23, MINN. JANDRO EQUIPMENT CO.

PERMA does hereby appoint JANDRO EQUIPMENT CO. as the Distributor of its "Perma Anti-Skid Control" and the Distributor accepts such appointment upon the following terms and conditions:

1. The area and territory for said distributorship shall be NORTH DAKOTA - SOUTH DAKOTA MINNESOTA, NORTHERN WISCONSIN (SEE MAP) UPPER PENINSULAR MICHIGAN - (SEE MAP) as outlined in the cross-hatched portion of the marketing maps attached to this agreement.

2. The Distributor shall employ the necessary sales personnel and service people to effectively merchandise the "Perma Anti-Skid Control" for said areas and territories which are the subject of said distributorship, as set forth in Paragraph 1.

3. The Distributor shall keep on hand and maintain a reasonable inventory of "Perma Anti-Skid Controls" and all parts incidental and necessary for the sale, repair, maintenance and installation of the same. After the expiration of six (6) months from the date hereof, a reasonable inventory shall be deemed to mean a thirty (30) day supply based on past experience.

4. The Distributor shall at all times employ a qualified mechanic to properly service and train other service and installation men. Said Qualified mechanic shall be trained by PERMA at PERMA'S main office in North Attleboro, Massachusetts, at the sole cost and expense of the Distributor, including travel, maintenance and salary of the employees as personnel. PERMA, however, shall make no training charge thereof.

5. The Distributor shall conform to standards and technical [illegible] from time to time issued by Perma with respect to [illegible]

(paragraphs 4 and 5 transcribed on page E-330A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last two paragraphs thereof read as follows:

> "4. The Distributor shall at all times employ a qualified mechanic to properly service and train other service and installation men. Said qualified mechanic shall be trained by PERMA at PERMA's main office in NORTH ATTLEBORO, MASSACHUSETTS, at the sole cost and expense of the Distributor, including travel, maintenance, and salary of its employees or personnel. PERMA, however, shall make no training charge thereof.
>
> "5. The Distributor shall conform to standards and technical (?) from time to time (?) by PERMA with regard to (installing?) . . . "

tics, service, maintenance, repair and replacement of "Perma Anti-Skid Controls" which are subject to this agreement, and to only use repair or replacement parts for such units which have been supplied or approved in writing by PERMA.

6. The Distributor shall, monthly and on such other times as PERMA may reasonably request, furnish PERMA reports of sales, inventory on hand and any other pertinent information on forms supplied by PERMA.

7. It is the intent of this Agreement that no products supplied by PERMA to the Distributor shall be sold, distributed, marketed or otherwise disposed of by the Distributor outside of the area and territory designated in Paragraph 1, and the Distributor shall use every effort to accomplish said intent and will take all necessary action on its part to enforce this intent in its dealings with its agents, dealers and customers in general.

8. The Distributor shall use its best efforts through its organization to distribute and procure sales, and create a demand for said "Perma Anti-Skid Control".

9. This Agreement is personal between PERMA and the Distributor, and shall not be assignable without the prior written consent of PERMA.

10. During the term of this Agreement or any extension thereof, the Distributor shall not market or otherwise deal with any other product or products which are competitive to the "Perma Anti-Skid Control".

11. The Distributor acknowledges that certain phases of the manufacture of the "Perma Anti-Skid Control" are outside the control of PERMA, and PERMA shall not be responsible for any delays in shipping said product for reasons beyond its control, including but not limited to strikes, fire, flood or other acts of God. PERMA, however, shall keep the Distributor periodically advised of the availability of the product for shipment.

12. The Distributorship herein granted shall be exclusive in the area and territory set forth in Paragraph 1, provided however, that PERMA reserves the right to make direct sales to national fleet accounts, original equipment manufacturers (such as automobile, truck and power vehicle manufacturers), manufacturers who supply components to the Original Equipment Manufacturers Market, U. S. Government Agencies, and national chain stores as herein used shall be defined as o

(paragraph 12 transcribed on page E-331A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last paragraph thereof reads as follows:

> "12. The Distributorship herein granted shall be exclusive in the area and territory set forth in Paragraph 1, provided however, that PERMA reserves the right to make direct sales to national fleet (?), original equipment manufacturers (such as automobile, truck and power vehicle manufacturers), manufacturers of the supply components to the Original Equipment Manufacturers Market, U.S. Government Agencies, and national (?)..., National Chain Stores as herein used shall be defined as a ...."

chain having ten or more retail outlets. In the event of any direct sales by PERMA to a National Chain Store, for re-distribution to branch stores, PERMA will ask for and use every reasonable effort to obtain the National Chain Stores Main Offices' co-operation in determining the units shipped to their branches as outlined in the territory described in Paragraph 1. Upon receipt of this information, PERMA shall pay the distributor 5% of the net billing price for those units that were trans-shipped into his territory. Said 5% re-imbursement shall be considered voluntary on the part of PERMA, and Distributor agrees that PERMA shall not be held liable if it is impossible or impracticable to obtain the necessary paper work from the National Chain Store, to issue said 5% over-ride.

13. PERMA shall supply literature and other sales data, which it may from time to time issue, to the Distributor on a cost plus handling basis, and shall furnish the Distributor with such other information as, in the opinion of PERMA, may be useful to the Distributor in the sale of "Perma Anti-Skid Controls".

14. PERMA shall refer to the Distributor all directly received inquiries and orders for "Perma Anti-Skid Controls", involving sales within the area and territory set forth in Paragraph 1 of this Agreement, with those exceptions as outlined in Paragraph 12 supra.

15. The Distributor shall accept separate billing of $1.85 each, net for each complete "Perma Anti-Skid Control" shipped to it; and PERMA shall contribute an equal amount each. Said total amount of $3.70 for each complete "Perma Anti-Skid Control" shall be set up as an advertising fund to be spent by the Distributor in accordance with PERMA'S advertising policy.

16. The term of this Distributorship shall be for the period from the date of this Agreement to APRIL 2 - 1965. The Distributor shall have an option to renew this Agreement for 5 successive annual terms, provided that (a) he shall have kept and performed all the terms and conditions of this Agreement, to be kept and performed by the Distributor during the preceding term; and (b) in the preceding term shall have ordered a minimum of 17,000 - Seventeen Thousand "Perma Anti-Skid Controls" or the average "Perma Anti-Skid Controls" per 1,000

of registered automobiles purchased by all other Distributors in their respective marketing areas whichever is greater. Average as herein used shall mean the total sales by all PEMA Distributors divided by the total number of automobiles registered in these Distributors' areas. Said options shall be exercised by United States Registered Mail addressed to PEMA at least sixty (60) days before the expiration of the prior term.

In case of failure on the part of the Distributor to pay for all merchandise sold to it by PEMA promptly when the same becomes due and payable, or in case the Distributor shall neglect or fail to perform or observe any of the terms and conditions imposed upon the Distributor by this Agreement, or in the event that the Distributor makes an assignment for the benefit of Creditors or is adjudged a bankrupt or a debtor, or reorganization, arrangement, or similar petition or proceedings be filed by or against it under any chapter or provision of the Federal Bankruptcy Act, so-called, or in the event a Receiver is appointed over the assets of said Distributor and said receiver is not vacated within ten (10) days thereafter; then and in any of the above events, PEMA shall at its option, and not withstanding any waiver of any prior breach of any covenant, term or condition, terminate this Agreement upon written notice to that effect addressed to the Distributor by United States Registered Mail, provided however, if the breach of any term or condition of this Agreement is of such nature that it may be cured or remedied, the Distributor shall have a period of thirty (30) days from the receipt of notice of said breach from PEMA to cure or remedy said default.

20. In the event of the termination of this Agreement for any cause whatsoever, PEMA shall have the right to purchase all its products in the possession of the Distributor at the then prevailing price therefor, minus a 10% handling charge.

19. (c) The Distributor does hereby order on the date hereof in payment from time to time PEMA a total of Seventeen Thousand (17,000) Complete "Steam Assembled Cabriolet" for the period commencing with April 1-1949 and ending on April 1-1950. The delivery schedule shall be as follows:

(paragraphs 18, 19 transcribed on page E-333A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last two paragraphs thereof read as follows:

> "18. In the event of the termination of this Agreement for any cause whatsoever, PERMA shall have the right to purchase all its (products?) in the possession of the Distributor at the then prevailing price therefore, \minus?) a (?) handling charge.
>
> 19. (a) The Distributor does hereby order on the date hereof in proper (form?) (?)... a total of SEVENTEEN THOUSAND (17,000) (?) 'Perma Anti-Skid Controls' for the period starting with APRIL 2, 1964 and ending on APRIL 1, 1965. The delivery schedule shall be as follows:"

[PX 27, p 5]

300- APRIL 1964 —
500- MAY 1964 —
750- JUNE 1964 —
750- JULY 1964 —

1000- AUGUST-1964
1500- SEPTEMBER 1964
2000- OCTOBER 1964
2100- NOVEMBER 1964
2100- DECEMBER 1964
2000 MARCH 1965

(b) The cost to the Distributor for the complete "Perma Anti-Skid Control" shall be Forty-six dollars and 27/100 ($46.27) plus three dollars and 70/100 ($3.70) Federal Excise Tax for each "Perma Anti-Skid Control" with freight F.O.B. North Attleboro, or Worcester, Massachusetts, U.S.A., it being specifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by PERMA to the Distributor. In order that the Distributor may have a flexible inventory we shall package Static Kits, Variable Kits, and Adapter Kits separately.

1-Complete "Perma Anti-Skid Control" consists of:

1 - each Static Kit, Variable Kit, Adapter Kit.

(c) The variable kits and/or adapter kits portion is packaged separately and of necessity must be custom made to fit each automobile. Distributors may purchase additional quantities of variable kits and/or adapter kits as a separate item in order that their inventory of "Perma Anti-Skid Controls" be flexible at all times. Perma Research and Development Company shall exchange Variable Kits and/or Adapter Kits (in original packages) for Distributors (every 3 months) to keep their inventory under control and up to date at all times. There will be no charge for this exchange policy. The only expense to the Distributor will be transportation charges both ways.

(d) Payments to PERMA by the Distributor for Product shall be as follows:

PAYMENT-IN-ADVANCE - APRIL SHIPMENT - 1% C.D.
PAYMENT-IN-ADVANCE - MAY SHIPMENT - 1% C.D.
PAYMENT-IN-ADVANCE - JUNE SHIPMENT - 1% C.D.

Subsequent shipments shall be on regular terms of 1% - Cash Discount 10 Days - 30 DAYS NET.

IN WITNESS WHEREOF, PERMA RESEARCH AND DEVELOPMENT COMPANY has caused these presents to be signed and its corporate seal to be hereunto affixed by Clinton T. Crellin, its Secretary.

(last paragraph transcribed on page E-334A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last paragraph thereof reads as follows:

> "IN WITNESS WHEREOF, PERMA RESEARCH AND DEVELOPMENT COMPANY has (?)... to be signed and its corporate seal to be (?) affixed by Clinton T. Crolius, its Secretary."

purpose duly authorized, and the Board of Directors has caused these presents to be signed and its corporate seal to be hereunto affixed by Clinton T. Crolius, its Secretary for this purpose duly authorized, this 9th day of March, A. D. 1964.

FERRO RESEARCH AND DEVELOPMENT COMPANY

By: Samuel Kaufman

V. P. Marketing

Sandro Equipment Co., Inc.

By: Clinton T. Crolius

President

Corporate Trust

Title Trust

% Sale ?

PLAINTIFF'S EXHIBIT 28 [PX 28, p 1].

MANUFACTURER'S AGENT AGREEMENT

THIS AGREEMENT, made and entered into as of the 4th day of September, 1964, by and between THOMAS H. COULTER, now of Des Moines, Polk County, Iowa, hereinafter called "Coulter", and PERMA RESEARCH AND DEVELOPMENT COMPANY, a Delaware Corporation with its principal place of business in North Attleboro, Massachusetts, hereinafter called "Perma", as follows:

WHEREAS, Perma has developed and is producing a product known as "Perma Anti-Skid Control" and

WHEREAS, Coulter has performed and will perform valuable services in developing nationwide markets for the said product and,

WHEREAS, Perma desires that Coulter continue to perform services in developing and servicing such nationwide markets, and

WHEREAS, Coulter is willing to perform such services on the terms and conditions hereinafter set forth;

NOW, THEREFORE, it is mutually agreed as follows:

1. Perma hereby engages Coulter as a manufacturer's agent, and Coulter does hereby agree to perform or cause to be performed the functions of such manufacturer's agent, as herein set forth.

2. Coulter shall by himself and through others continue in his efforts to develop a market for Perma's product with major oil companies (which shall include oil companies operating throughout the United States), regional oil companies (which shall include oil companies operating in more than one State of the United States), and certain suppliers to such oil companies commonly known as tire, battery and accessory suppliers or "TBA" suppliers (such as Goodyear, Firestone and the like), by sales to such major and regional oil companies and TBA suppliers for resale to consumers through filling stations and other retail outlets. Coulter undertakes and agrees to sell to and service such customers and maintain the required sales personnel to contact and train the customers personnel. Coulter also agrees to maintain the necessary technical staff to train customers personnel and trouble shoot Perma products in the field, (Perma agrees to provide major engineering services requested and needed by such customers in the event of major engineering problems.)

E-336

3. As full compensation for such services, (and any expenses incurred in performing them), Perma shall pay and Coulter shall accept a commission equal to 10% of the net billing price by Perma on orders originated by Coulter in accounts covered by paragraph two of this agreement. Billing price on which commission shall be paid shall not include -

Federal Excise Tax
State Taxes
Freight
Cash Discounts

No commissions shall be paid to Coulter for the sale of any sales promotional material. It is further agreed that the commission rate shall be reviewed six (6) months from the above date to determine if an adjustment is to be made. Both parties agree that if an adjustment in the commission rate become necessary, then said adjustment shall be arrived at by mutual agreement.

4. The commissions herein provided shall be paid to Coulter or to his order not later than the 15th of the month following the receipt of payment by Perma from those orders covered by this agreement. Coulter shall have the right, by himself or by his representatives, to inspect the books of Perma insofar as the said books shall pertain to the sales to the customers herein described, such inspection to be at the principal office of Perma, during regular business hours, and at no more than reasonable intervals.

5. Nothing herein contained shall be construed as constituting Coulter an employee of Perma, and Coulter shall not have the authority by reason hereof to incur any expenses on behalf of Perma, accept any orders on behalf of Perma, or otherwise act for Perma except as herein specified. Coulter further agrees that he will adhere to Perma's sale policies and pricing schedules that may be issued from time to time.

6. It is understood by the parties that Coulter is contemplating the formation of a corporation. If such corporation is so formed, or if Coulter shall acquire a corporation now existing, it is expressly understood that this agreement may be assigned by Coulter to such corporation, and that such corporation, and that such corporation shall succeed to all of the rights, duties, and obligations of Coulter hereunder. Provided however,

that Coulter shall be and remain a majority stockholder in such corporation, and active in the management thereof.

7. The term of this agreement shall be for the period from the date of this agreement to August 31, 1965. Coulter shall have the option to renew this agreement for four (4) successive annual terms and provided that--

(a) Coulter shall have kept and performed all the terms and conditions of this agreement during the preceding term and -

(b) Coulter shall have sold and Perma shall have delivered and received payment for shipments totaling not less $1,000,000. during the preceding year.

In the event of non-compliance or violation with any protion of this agreement and such violation is not corrected within thirty (30) days after written notification then Perma shall have the right to terminate this agreement upon written notice to Coulter by United States Registered Mail. Said termination shall become effective ten (10) days from the date of said termination notice.

8. Perma agrees that it shall use its best efforts to assign this agreement in the event there shall be a successor or merge of Perma Research and Development Company.

9. This contract shall be construed in accordance with the laws of Massachusetts.

IN WITNESS WHEREOF, these parties have hereunto set their hand on the day and year first above written.

Thomas H. Coulter

PERMA RESEARCH AND DEVELOPMENT COMPANY

BY:

PRICE LIST FOR TOM COULTER REPRESENTATIVE AGREEMENT

T. B. A.

Retail-$129.95
- 30%
Dealer 91.00

Jobber 77.06 - minus 5% warehousing = 73.11

69.97
- Less 5% master discount
66.49
1.66 - Co-op Adv. rebate
64.83 - Base recovery to Perma.

Direct Purchase & Sale to Oil Companies

$129.95
91.00
77.06 - 3.89 (5% warehousing) = $73.20

65.00
- Adv. & Prom. allowances they can earn on escalator basis
59.85

PLAINTIFF'S EXHIBIT 29

BETWEEN:

PERMA RESEARCH AND DEVELOPMENT COMPANY (Inc), a Delaware Corporation, doing business at 345 East Washington Street, North Attleboro, Massachusetts, U.S.A. hereinafter called PERMA

- and -

CLARELEN IMPORTERS LIMITED, a corporation duly organized under the laws of the Province of Saskatchewan, Canada with its Head Office at 220 Mitchell Block, Prince Albert, Saskatchewan, Canada, hereinafter called CLARELEN.

WITNESSETH:

Whereas PERMA is engaged in the sale of an Anti-Skid device known as "Perma Anti-Skid" and

Whereas CLARELEN is engaged in the business of maintaining a distributorship for the sale and installation of automotive products, and

Whereas PERMA and CLARELEN desire to enter into an agreement for the sale of Perma's Anti-Skid device to users, wholesalers and retailers on an exclusive basis for the Dominion of Canada by CLARELEN:

Now, therefore, in consideration of the mutual covenant and agreements herein contained, PERMA does hereby appoint in said territory (Dominion of Canada) CLARELEN in the following terms and conditions:

1. CLARELEN agrees to employ the necessary salespeople to effectively merchandise the "Perma Anti-Skid unit for the Dominion of Canada and it is expressly agreed that in the event minimum orders are not placed by CLARELEN and payment received by Perma as hereinafter set forth, it shall be deemed that CLARELEN has not complied with this section and this agreement shall become null and void at the sole option of PERMA.

2. CLARELEN shall keep on hand and maintain a reasonable inventory of Perma Anti-Skid units and all parts incidental and necessary for the sale, repair, maintenance and installation of same.

3. CLARELEN agrees to conform to standards and technical instructions from time to time issued by PERMA with respect to sales, advertising, promotion, installation, service, maintenance repair and replacement of Perma Anti-Skid units which are subject to this agreement, and to only use repair or replacement parts for such units which have been supplied or approved in writing by PERMA.

4. CLARELEN shall keep PERMA advised periodically as to its activities in connection with sales or solicitation of sales of the Perma Anti-Skid units and more specifically will submit to PERMA monthly, written reports showing movement, inventory on hand and any and all other reasonable information that PERMA may request.

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5. CLARELEN shall not contract obligations in the name of PERMA, and it will not make any representation, guaranty, or warranty with respect to said PERMA Anti-Skid units, except such as is authorized by PERMA.

6. CLARELEN warrants and guarantees that all product received from PERMA will be sold, distributed or otherwise marketed in the Dominion of Canada and will use every effort, legal or otherwise to accomplish the intent of this section which is that no product supplied to CLARELEN by PERMA shall be sold, distributed, marketed or otherwise disposed of outside the Dominion of Canada and that it will take all necessary action on its part to enforce this intent in its dealings with its agents, distributors and customers in general.

7. CLARELEN agrees that no arrangements or agreements shall be made with any of its agents, distributors, or customers in general inconsistant with this agreement.

8. CLARELEN will ship and invoice at its own expense, and will assume credit risks in its own customers.

9. CLARELEN shall not have the right to assign this agreement or distributorship to any person, firm or corporation without the written consent of PERMA.

10. CLARELEN agrees that during the continuance of this agreement it will not market or otherwise deal with any other product or products which may be deemed to be in conflict with the marketing of the Perma Anti-Skid unit.

11. PERMA and CLARELEN agree that best efforts shall be expended to comply with the terms of this agreement excepting that it is hereby acknowledged by CLARELEN that certain phases of the manufacture of the Perma Anti-Skid unit is outside the control of PERMA and PERMA shall not be held accountable for delays in shipping product for reasons beyond its control which shall include but not be limited thereto, strikes, fire, flood or other acts of God, however in any event PERMA shall keep CLARELEN advised periodically of the availability of product for shipment.

12. PERMA agrees that CLARELEN shall be the exclusive importer and distributor of the PERMA Anti-Skid unit for the Dominion of Canada, however it is specifically agreed that PERMA reserves the right to make sales to original equipment manufacturers such as automobile, truck and power vehicle manufacturers, United States Government agencies and national accounts, however, PERMA shall use reasonable effort to refer to CLARELEN all purchases of Perma Anti-Skid units by the foregoing accounts for consumption or sale in the Dominion of Canada, however it is specifically agreed that in no event shall PERMA be held accountable or denied the right or be deemed to have relinquished any rights for sale or to sell any of its products to the above type of accounts and it is further specifically agreed that CLARELEN shall not under any circumstances sell or arrange to sell any of the above type accounts, whose principal offices or place of business is located outside the Dominion of Canada.

13. PERMA agrees to supply literature and other sales data, which it may from time to time issue, to CLARKLEN on a cost plus handling basis, and to furnish CLARKLEN with such other information as, in the opinion of PERMA, may be useful to CLARKLEN in the sale of Perma Anti-Skid units.

14. PERMA agrees to refer to CLARKLEN all directly received inquiries and orders for Perma Anti-Skid units, involving sales within the Province of Canada.

15. It is hereby agreed that CLARKLEN, shall, on the date hereof, be deemed to have ordered in proper form from PERMA, sixteen thousand (16,000) " Perma Anti-Skid" units with delivery to be made as follows:

(a) Four thousand (4,000) Perma Anti-Skid units to be shipped on or before September 30, 1963 with shipments to start in minimum lots of 100 units, on or before July 1, 1963.

(b) Four Thousand (4,000) Perma Anti-Skid units to be delivered in each subsequent quarterly period (3 months) at an average number of not less than 1300 units per month (S.K. LL

(c) The cost to CLARELEN shall be Forty six dollars and 27/100ths ($46.27) for each Perma Anti-Skid unit on a net basis with freight F.O.B. North Attleboro, Massachusetts, U.S.A., it being specifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by PERMA to CLARELEN.

(d) All payments by CLARELEN to PERMA for product shall be payments in advance for product to be shipped and shall be paid for prior to shipment in cash, bank draft or certified check in United States funds.

(e) Upon the signing of this agreement CLARELEN shall make an initial payment to PERMA in an amount of Forty Thousand ($40,000) dollars which amount shall be an advance payment for the final and last units to be shipped of the above mentioned order of Sixteen thousand (16,000) units.

(f) In lieu of interest on the Forty Thousand ($40,000) dollar deposit mentioned in paragraph (e) hereof, PERMA hereby agrees to make a gratis shipment of fifty-two units upon completion of this contract.

16. PERMA hereby agrees that if CLARELEN purchases from them Sixteen Thousand (16,000) Perma Anti-Skid units during the continuance of this present contract, CLARELEN shall remain the exclusive distributor of Perma Anti-Skid Units in the Dominion of Canada for a further period of one year commencing on the 1st day of Jul., A.D. 1964.

17. PERMA h reby further agrees that the Forty Thousand ($40,000) deposit paid under the terms of the contract, is also consideration on the part of CLARELEN to remain the exclusive distributor in Canada of the Perma Anti-Skid units during the term of this contract and any renewal term and PERMA further agrees that should they knowingly (SR) (ZR) allow or tolerate any other person to distribute Perma Anti-Skid Units in Canada, during the continuance of this Contract or any renewal thereof then, and in that event, they shall return to CLARELEN immediately, the Forty Thousand ($40,000 dollar paid as a deposit on this contract, such payment to be in cash and not in kind.

18. It is hereby agreed by the parties hereto that title to any product subject to this agreement shall remain in PERMA until paid for by CLARELEN as set forth herein and under this agreement. No product shall be returned without prior written approval of PERMA and PERMA reserves the right to accept in original cartons free from damage, product returns at a 15% handling charge.

19. In the event there is any breach of the covenants and terms contained herein by CLARELEN, the said CLARELEN agrees that the said Forty Thousand ($40,000) dollar advance payment for last product herein referred to, shall be retained by PERMA as liquidated damages with no obligation on the part of PERMA to proceed further in accordance with the terms hereof.

20. PERMA hereby agrees that the Anti-Skid units supplied by them to CLARELEN shall function so as to accomplish properly the purposes for which they were designed and sold and PERMA hereby agrees to replace any unit supplied to CLARELEN which does not function properly or not built to standard specifications and PERMA further agrees to protect, indemnify and save harmless CLARELEN from any liabilities which it might incur by reason of malfunctioning of the Perma Anti-Skid units, by reason of defects in the Perma Anti-Skid Units or by reason of the Perma Anti-Skid

units not accomplishing the purpose for which it was designed and sold providing said unit is installed in accordance with the instructions provided by Pama. SKD

IN WITNESS WHEREOF said PERMA RESEARCH AND DEVELOPMENT COMPANY and CLARELEN IMPORTERS LIMITED have caused their presents to be signed and their respective corporate seals to be hereunto affixed by their respective officers thereunto duly authorized on the date and year first above mentioned.

PERMA RESEARCH AND DEVELOPMENT COMPANY

BY: Samuel Karp
Samuel Karp, [illegible] Manager.

CLARELEN IMPORTERS LIMITED

BY: Clarence Kurtenbach
Clarence Kurtenbach, President.

BY: Leonard Kurtenbach
Leonard Kurtenbach, General Sales Manager.

CLARELEN IMPORTERS LIMITED Seal

It is further agreed that Rama Research and Development Co. will propose a plan, on or before Sept. 30. 1963, for the manufacture of Rama Research and Development Co's products under a licensing agreement and/or royalty basis for the Dominion of Canada.

Such plan may be accepted or rejected by Clarion Industries Ltd. within 45 days after such proposal is presented to Clarion Industries Ltd. If Clarion Industries Ltd shall accept the above manufacturing licensing agreement as proposed by Rama Research and Development Co. this agreement dated June 10th 1963 shall become null and void, However Rama Research and Development Co. agrees to deliver 40,000 Rama Semi-Shell Units as outlined in paragraph 15 section 2 above and to be delivered in accordance with the schedule as outlined in paragraph 15 section 2 above. In the event that Clarion Industries Ltd rejects the proposal of the manufacturing rights the contract

June 10, 1963
June 10th 1963

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(PX 29. p 5)

DATE: Aug. 4, 1964

[PX 30A, p 1]

PLAINTIFF'S EXHIBIT 30A

A G R E E M E N T

AGREEMENT MADE by and between PERMA RESEARCH AND DEVELOPMENT COMPANY, a Delaware Corporation with its principal place of business in North Attleboro, Commonwealth of Massachusetts, hereinafter referred to as "PERMA" and

MONITOR ENTERPRISES, INC.

PERMA does hereby appoint

as the Distributor of its "Perma Anti-Skid Control" and the Distributor accepts such appointment upon the following terms and conditions:

1. The area and territory for said distributorship shall be Northern New Jersey Marketing Area as outlined in the unshadowed portion of the marketing maps attached to this agreement.

2. The distributor hereby agrees that he will employ at all times a minimum of six (6) trained sales people and one (1) trained technician to effectively merchandise the "Perma Anti-Skid Control" for said areas and territories, which are the subject of said distributorship as set forth in Paragraph 1. The Distributor agrees said sales people shall be hired and trained at a minimum rate of two (2) per month.

3. The Distributor shall keep on hand and maintain a reasonable inventory of "Perma Anti-Skid Controls" and all parts incidental and necessary for the sale, repair, maintenance and installation of the same. After the expiration of six (6) months from the date hereof, a reasonable inventory shall be deemed to mean a thirty (30) day supply based on past experience.

4. The Distributor shall at all times employ a qualified mechanic to properly service and train other service and installation men. Said qualified mechanic shall be trained by PERMA at PERMA'S main office in NORTH ATTLEBORO, MASSACHUSETTS, at the sole cost and expense of the Distributor, including travel, maintenance and salary of its employees or personnel. PERMA, however, shall make no training charge thereof.

5. The distributor shall conform to standards and technical specifications from time to time issued by Perma with respect to install...

n, service, maintenance, repair and replacement of "Perma Anti-Skid ntrols" which are subject to this agreement, and to only use repair or placement parts for such units which have been supplied or approved in iting by PERMA.

The Distributor shall, monthly and on such other times as PERMA y reasonably request, furnish PERMA reports of sales, inventory on hand d any other pertinent information on forms supplied by PERMA.

It is the intent of this Agreement that no products supplied by RMA to the Distributor shall be sold, distributed, marketed or otherwise sposed of by the Distributor outside of the area and territory designated Paragraph 1, and the Distributor shall use every effort to accomplish id intent and will take all necessary action on its part to enforce this tent in its dealings with its agents, dealers and customers in general.

The Distributor shall use its best efforts through its organi- tion to distribute and procure sales, and create a demand for said rma Anti-Skid Control".

9. This Agreement is personal between PERMA and the Distributor, and shall not be assignable without the prior written consent of PERMA.

10. During the term of this Agreement or any extention thereof, the Distributor shall not market or otherwise deal with any other product or products which are competitive to the "Perma Anti-Skid Control".

11. The Distributor acknowledges that certain phases of the manufacture of the "Perma Anti-Skid Control" are outside the control of PERMA, and PERMA shall not be responsible for any delays in shipping said product for reasons beyond its control, including but not limited to strikes, fire, flood or other acts of God. PERMA, however, shall keep the Distributor periodically advised of the availability of the product for shipment.

12. The Distributorship herein granted shall be exclusive in the area and territory set forth in Paragraph 1, provided however, that PERMA reserves the right to make direct sales to national fleet accounts, original equipment manufacturers (such as automobile, truck and power vehicle manufacturers), manufacturers who supply components to the Original Equipment Manufacturers Market, U. S. Government Agencies, and national chain stores. National Chain Stores as herein used shall be defined as a

Chain having ten or more retail outlets. In the event of any direct sales by PERMA to a National Chain Store, for re-distribution to branch stores, PERMA will ask for and use every reasonable effort to obtain the National Chain Stores Main Offices' co-operation in determining the units shipped to their branches as outlined in the territory described in Paragraph 1. Upon receipt of this information, PERMA shall pay the distributor 5% of the net billing price for those units that were trans-shipped into his territory. Said 5% re-imbursement shall be considered voluntary on the part of PERMA, and Distributor agrees that PERMA shall not be held liable if it is impossible or impracticable to obtain the necessary paper work from the National Chain Store, to issue said 5% over-ride.

13. PERMA shall supply literature and other sales data, which it may from time to time issue, to the Distributor on a cost plus handling basis, and shall furnish the Distributor with such other information as, in the opinion of PERMA, may be useful to the Distributor in the sale of "Perma Anti-Skid Controls".

14. PERMA shall refer to the Distributor all directly received inquiries and orders for "Perma Anti-Skid Controls", involving sales within the area and territory set forth in Paragraph 1 of this Agreement, with those exceptions as outlined in Paragraph 12 supra.

15. The Distributor shall accept separate billing of $1.85 each, net for each complete "Perma Anti-Skid Control" shipped to it; and PERMA shall contribute an equal amount each. Said total amount of $3.70 for each complete "Perma Anti-Skid Control" shall be set up as an advertising fund to be spent by the Distributor in accordance with PERMA'S advertising policy.

16. The term of this Distributorship shall be for the period from the date of this Agreement to July 31, 1965. The Distributor shall have an option to renew this Agreement for 5 successive annual terms, provided that (a) it shall have kept and performed all the terms and conditions of this Agreement, to be kept and performed by the Distributor during the preceding term; and (b) in the preceding term shall have ordered a minimum of Nine Thousand (9,000) "Perma Anti-Skid Controls" or the average "Perma Anti-Skid Controls" per 1,000

of registered automobiles purchased by all other Distributors in their respective marketing areas whichever is greater. Average as herein used shall mean the total sales by all PERMA Distributors divided by the total number of automobiles registered in those Distributors' areas. Said options shall be exercised by United States Registered Mail addressed to PERMA at least sixty (60) days before the expiration of the prior term.

17. In case of failure on the part of the Distributor to pay for all merchandise sold to it by PERMA promptly when the same becomes due and payable, or in case the Distributor shall neglect or fail to perform or observe any of the terms and conditions imposed upon the Distributor by this Agreement, or in the event that the Distributor makes an assignment for the benefit of Creditors or is adjudged a bankrupt or a debtor, re-organization, arrangement, or similar petition or proceedings be filed by or against it under any chapter or provision of the Federal Bankruptcy Act so-called, or in the event a Receiver is appointed over the assets of said Distributor and said receiver is not vacated within ten (10) days thereafter; then and in any of the above events, PERMA shall at its option, and not withstanding any waiver of any prior breach of any covenant, term or condition, terminate this Agreement upon written notice to that effect addressed to the Distributor by United States Registered Mail, provided however, if the breach of any term or condition of this Agreement is of such nature that it may be cured or remedied, the Distributor shall have a period of thirty (30) days from the receipt of notice of said breach from PERMA to cure or remedy said default.

18. In the event of the termination of this Agreement for any cause whatsoever, PERMA shall have the right to purchase all its products in the possession of the Distributor at the then prevailing price therefore, minus a 15% handling charge.

19. (a) The Distributor does hereby order on the date hereof in proper form from PERMA a total of Nine Thousand (9,000) Complete "Perma Anti-Skid Controls" for the period starting with August 15, 1964 and ending on July 31, 1965 The delivery schedule shall be as follows:

Week of August 7th - 250
on or about Sept 15 - 500
" " Oct 15 - 500
" " Nov 15 - 500
" " Dec 15 - 1000

on or about Jan 15 - 1965 - 1000
" " Feb 15 - 1965 - 1000
" " Mar 15 - 1965 - 1000
" " April 15 - 1965 - 1000
" " May 15 - 1965 - 750
" " June 15 - 1965 - 750
" " July 15 - 1965 - 750

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(b) The cost to the Distributor for the complete "Perma Anti-Skid Control" shall be in accordance with that price list as may be issued by Perma Research and Development Company and identified as Warehouse Distributor or Area Distributor price list, with freight F.O.B. North Attleboro, or Elizabeth, New Jersey, U.S.A., it being spcifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by PERMA to the Distributor. In order that the Distributor may have a flexible inventory we shall package Static Kits, Variable Kits, and Adapter Kits separately.

1 - Complete "Perma Anti-Skid Control" consist of:

1 - each Static Kit, Variable Kit, Adapter Kit.

(c) The variable kits and/or adapter kits portion is packaged separately and of necessity must be custom made to fit each automobile. Distributors may purchase additional quantities of variable kits and/or adapter kits as a separate item in order that their inventory of "Perma Anti-Skid Controls" be flexible at all times. Perma Research and Development Company shall exchange Variable Kits and/or Adapter Kits (in original packages) for Distributors (every 3 months) to keep their inventory under control and up to date at all times. There will be no charge for this exchange policy. The only expense to the Distributor will be transportation charges both ways.

(d) Payments to PERMA by the Distributor for Product shall be as follows: Payment in advance for all shipments made prior to Dec 31-1964.
Terms as outlined on our Price Sheets
Thereafter

IN WITNESS WHEREOF, PERMA RESEARCH AND DEVELOPMENT COMPANY has caused these presents to be signed and its corporate seal to be hereunto affixed by [illegible] E-349

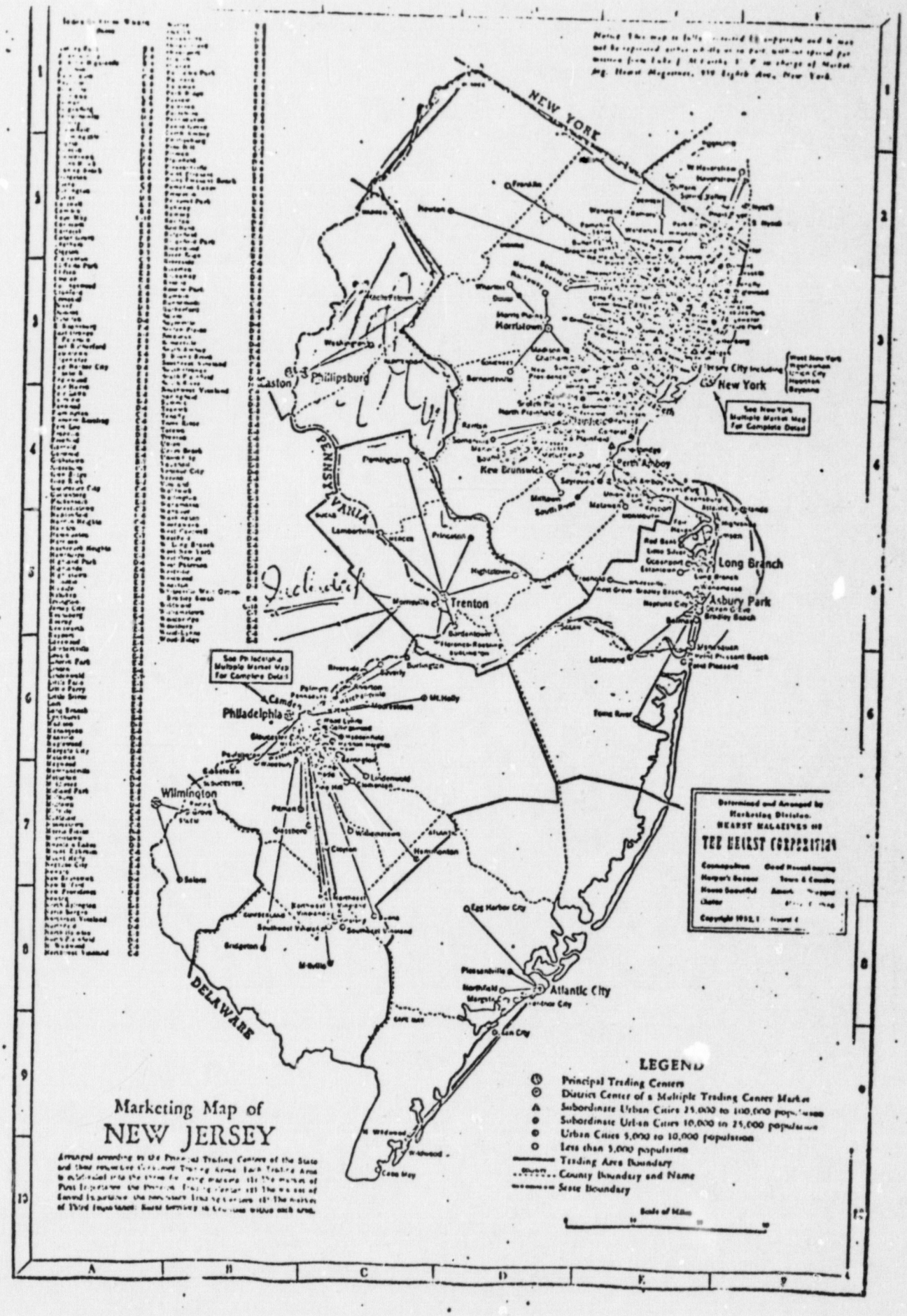
purpose duly authorized, and Board of directors has caused these presents to be signed and its corporate seal to be hereunto affixed by Clinton Colvin its Secretary for this purpose duly authorized, this 6th day of August, A.D. 1964.

PERMA RESEARCH AND DEVELOPMENT COMPANY

BY: J. A. Kenney (C) Vice Pres.

MONITOR ENTERPRISES, INC.

BY: Joseph Kelly V.P.


NEW YORK
Phillipsburg
Easton
Morristown
New York
See New York Multiple Market Map For Complete Detail
New Brunswick
Perth Amboy
Trenton
Long Branch
Asbury Park
See Philadelphia Multiple Market Map For Complete Detail
Philadelphia
Camden
Wilmington
Atlantic City
DELAWARE
Determined and Arranged by Marketing Division
HEARST MAGAZINES INC
THE HEARST CORPORATION
LEGEND
Principal Trading Centers
District Center of a Multiple Trading Center Market
Subordinate Urban Cities 25,000 to 100,000 population
Subordinate Urban Cities 10,000 to 25,000 population
Urban Cities 5,000 to 10,000 population
Less than 5,000 population
Trading Area Boundary
County Boundary and Name
State Boundary
Scale of Miles
Marketing Map of
NEW JERSEY

F.P. – Damages
Drive
PPE 16

Π 30A

 RECD NOV. 23 1964

PLAINTIFF'S EXHIBIT 30B

DATE: July 24, 1963

A G R E E M E N T

AGREEMENT MADE by and between PERMA RESEARCH AND DEVELOPMENT COMPANY, a Delaware Corporation with its principal place of business in North Attleboro, Commonwealth of Massachusetts, hereinafter referred to as "PERMA" and

Edward B. Wright, Jr.
Wright Bros. Greenhouse
2762 West Bancroft Street
Toledo, Ohio

PERMA does hereby appoint Edward B. Wright, Jr. as the Distributor of its "Perma Anti-Skid Control" and the Distributor accepts such appointment upon the following terms and conditions:

1. The area and territory for said distributorship shall be

THE STATE OF OHIO

as outlined in the unshadowed portion of the marketing maps attached to this agreement.

2. The distributor hereby agrees that he will employ at all times a minimum of six (6) trained sales people and one (1) trained technician to effectively merchandise the "Perma Anti-Skid Control" for said areas and territories, which are the subject of said distributorship as set forth in Paragraph 1. The Distributor agrees said sales people shall be hired and trained at a minimum rate of two (2) per month.

3. The Distributor shall keep on hand and maintain a reasonable inventory of "Perma Anti-Skid Controls" and all parts incidental and necessary for the sale, repair, maintenance and installation of the same. After the expiration of six (6) months from the date hereof, a reasonable inventory shall be deemed to mean a thirty (30) day supply based on past experience.

4. The Distributor shall at all times employ a qualified mechanic to properly service and train other service and installation men. Said qualified mechanic shall be trained by PERMA at PERMA'S main office in NORTH ATTLEBORO, MASSACHUSETTS, at the sole cost and expense of the distributor, including travel, maintenance and salary of its employees or personnel. PERMA, however, shall make no training charge thereof.

5. The distributor shall conform to standards and technical instructions from time to time issued by Perma with respect to installa--

tion, service, maintenance, repair and replacement of "Perma Anti-Skid Controls" which are subject to this agreement, and to only use re or replacement parts for such units which have been supplied or appr in writing by PERMA.

6. The Distributor shall, monthly and on such other times as PERMA may reasonably request, furnish PERMA reports of sales, inventory on hand and any other pertinent information on forms supplied by PERMA.

7. It is the intent of this Agreement that no products supplied by PERMA to the Distributor shall be sold, distributed, marketed or otherwise disposed of by the Distributor outside of the area and territory designated in Paragraph 1, and the Distributor shall use every effort to accomplish said intent and will take all necessary action on its part to enforce this intent in its dealings with its agents, dealers and customers in general.

8. The Distributor shall use its best efforts through its organization to distribute and procure sales, and create a demand for said "Perma Anti-Skid Control".

9. This Agreement is personal between PERMA and the Distributor, and shall not be assignable without the prior written consent of PERMA.

10. During the term of this Agreement or any extention thereof, the Distributor shall not market or otherwise deal with any other product or products which are competitive to the "Perma Anti-Skid Control".

11. The Distributor acknowledges that certain phases of the manufacture of the "Perma Anti-Skid Control" are outside the control of PERMA, and PERMA shall not be responsible for any delays in shipping said product for reasons beyond its control, including but not limited to strikes, fire, flood or other acts of God. PERMA, however, shall keep the Distributor periodically advised or the availability of the product for shipment.

12. The Distributorship herein granted shall be exclusive in the area and territory set forth in Paragraph 1, provided however, that PERMA reserves the right to make direct sales to national fleet accounts, original equipment manufacturers (such as automobile, truck and power vehicle manufacturers), manufacturers who supply components to the Original Equipment Manufacturers Market, U. S. Government Agencies, and national chain stores. National Chain Stores as herein used shall be defined as a

chain having ten or more retail outlets. In the event of any direct sales by PERMA to a National Chain Store, for re-distribution to branch stores, PERMA will ask for and use every reasonable effort to obtain the National Chain Stores Main Offices' co-operation in determining the units shipped to their branches as outlined in the territory described in Paragraph 1. Upon receipt of this information, PERMA shall pay the distributor 5% of the net billing price for those units that were trans-shipped into his territory. Said 5% re-imbursement shall be considered voluntary on the part of PERMA, and Distributor agrees that PERMA shall not be held liable if it is impossible or impracticable to obtain the necessary paper work from the National Chain Store, to issue said 5% over-ride.

13. PERMA shall supply literature and other sales data, which it may from time to time issue, to the Distributor on a cost plus handling basis, and shall furnish the Distributor with such other information as, in the opinion of PERMA, may be useful to the Distributor in the sale of "Perma Anti-Skid Controls".

14. PERMA shall refer to the Distributor all directly received inquiries and orders for "Perma Anti-Skid Controls", involving sales within the area and territory set forth in Paragraph 1 of this Agreement, with those exceptions as outlined in Paragraph 12 supra.

15. The Distributor shall accept separate billing of $1.85 each, net for each complete "Perma Anti-Skid Control" shipped to it; and PERMA shall contribute an equal amount each. Said total amount of $3.70 for each complete "Perma Anti-Skid Control" shall be set up as an advertising fund to be spent by the Distributor in accordance with PERMA'S advertising policy.

16. The term of this Distributorship shall be for the period from the date of this Agreement to July 31, 1965. The Distributor shall have an option to renew this Agreement for 5 successive annual terms, provided that (a) it shall have kept and performed all the terms and conditions of this Agreement, to be kept and performed by the Distributor during the preceding term; and (b) in the proceding term shall have ordered a minimum of fifteen thousand (15,000) "Perma Anti-Skid Controls" or the average "Perma Anti-Skid Controls" per 1,000

of registered automobiles purchased by all other Distributors in their respective marketing areas whichever is greater. Average as herein used shall mean the total sales by all PERMA Distributors divided by the total number of automobiles registered in those Distributors' areas. Said options shall be exercised by United States Registered Mail addressed to PERMA at least sixty (60) days before the expiration of the prior term.

17. In case of failure on the part of the Distributor to pay for all merchandise sold to it by PERMA promptly when the same becomes due and payable, or in case the Distributor shall neglect or fail to perform or observe any of the terms and conditions imposed upon the Distributor by this Agreement, or in the event that the Distributor makes an assignment for the benefit of Creditors or is adjudged a bankrupt or a debtor, re-organization, arrangement, or similar petition or proceedings be filed by or against it under any chapter or provision of the Federal Bankruptcy Act so-called, or in the event a Receiver is appointed over the assets of said Distributor and said receiver is not vacated within ten (10) days thereafter; then and in any of the above events, PERMA shall at its option, and not withstanding any waiver of any prior breach of any covenant, term or condition, terminate this Agreement upon written notice to that effect addressed to the Distributor by United States Registered Mail, provided however, if the breach of any term or condition of this Agreement is of such nature that it may be cured or remedied, the Distributor shall have a period of thirty (30) days from the receipt of notice of said breach from PERMA to cure or remedy said default.

18. In the event of the termination of this Agreement for any cause whatsoever, PERMA shall have the right to purchase all its products in the possession of the Distributor at the then prevailing price therefore, minus a 15% handling charge.

19. (a) The Distributor does hereby order on the date hereof in proper form from PERMA a total of Fifteen thousand (15,000) Complete "Perma Anti-Skid Controls" for the period starting with August 1, 1964 and ending on July 31, 1965. The delivery schedule shall be as follows:

500 -	1st week of August 1964	2,000 -	1st week of February 1965
0 -	" " " Sept. "	1,500 -	" " " March "
750 -	" " " Oct. "	1,500 -	" " " April "
1,500 -	" " " Nov. "	1,500 -	" " " May "
1,750 -	" " " Dec. "	1,000 -	" " " June "
2,000 -	" " " Jan. 1965	1,000 -	" " " July "

(b) The cost to the Distributor for the complete "Perma Anti-Skid Control" shall be in accordance with that price list as may be issued by Perma Research and Development Company and identified as Warehouse Distributor or Area Distributor price list, with freight F.O.B. North Attleboro, or Elizabeth, New Jersey, U.S.A., it being speifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by PERMA to the Distributor. In order that the Distributor may have a flexible inventory we shall package Static Kits, Variable Kits, and Adapter Kits separately.

1 - Complete "Perma Anti-Skid Control" consist of:

1 - each Static Kit, Variable Kit, Adapter Kit.

(c) The variable kits and/or adapter kits portion is packaged separately and of necessity must be custom made to fit each automobile. Distributors may purchase additional quantities of variable kits and/or adapter kits as a separate item in order that their inventory of "Perma Anti-Skid Controls" be flexible at all times. Perma Research and Development Company shall exchange Variable Kits and/or Adapter Kits (in original packages) for Distributors (every 3 months) to keep their inventory under control and up to date at all times. There will be no charge for this exchange policy. The only expense to the Distributor will be transportation charges both ways.

(d) Payments to PERMA by the Distributor for Product shall be as follows: Payment in advance - or Sight Draft Bill of Lading until December 31, 1964.

Open account in accordance with terms outlined on price sheets shall prevail starting with January Shipments.

IN WITNESS WHEREOF, PERMA RESEARCH AND DEVELOPMENT COMPANY has caused these presents to be signed and its corporate seal to be hereunto affixed by M.A. Cronin, its Secretary,

purpose duly authorized, and THE BOARD OF DIRECTORS has caused these presents to be signed and its corporate seal to be hereunto affixed by MR. CRAWNS, its SECRETARY, for this purpose duly authorized, this 24TH day of JULY, A.D. 1964.

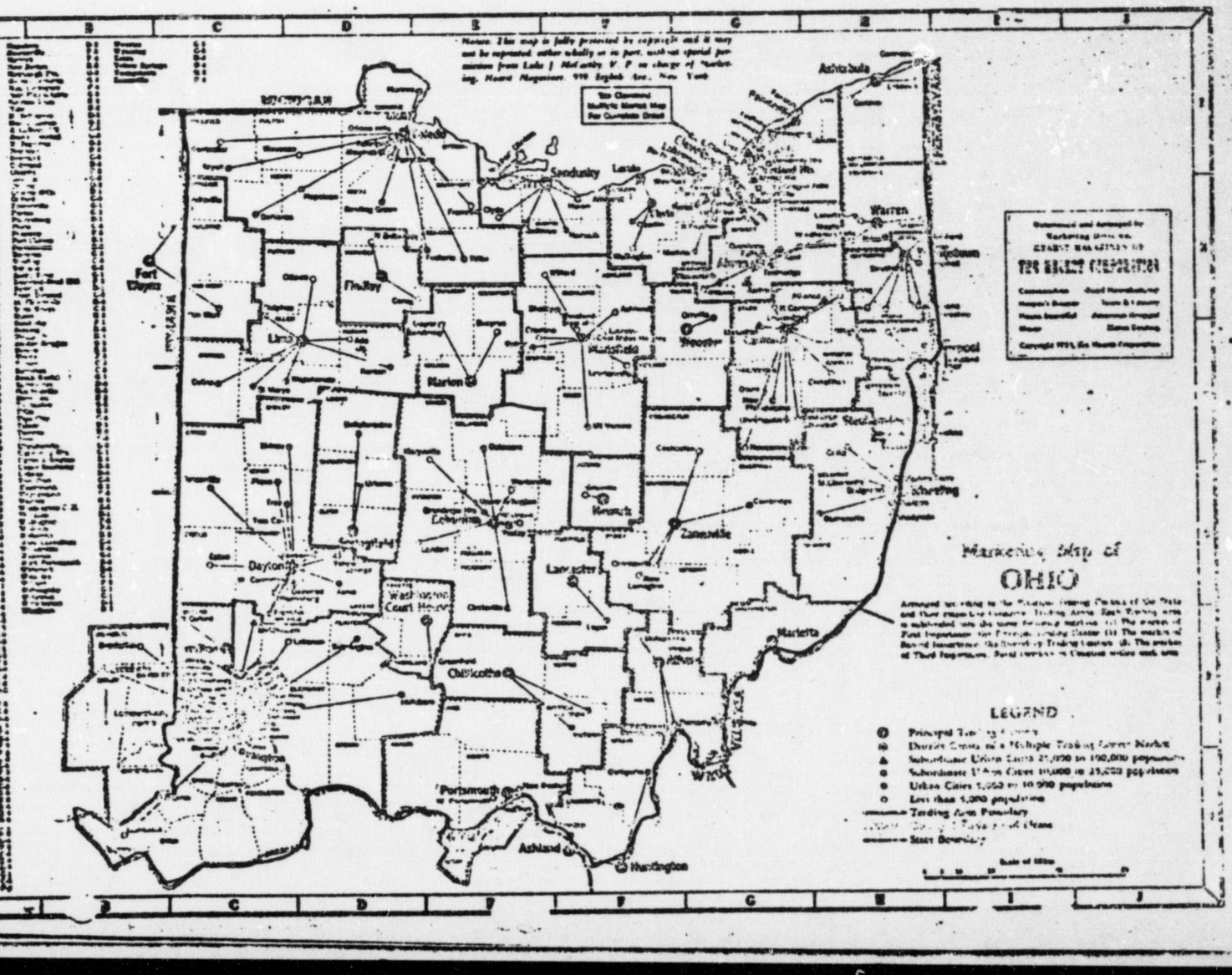
PERMA RESEARCH AND DEVELOPMENT COMPANY

BY: Perma Research + Devel. Corp (

By Samuel Kemp V.P.

BY: Car Controls, Inc.

by J.B. Wright Jr. Pres.


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MICHIGAN
INDIANA
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Lorain
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Akron
Canton
Youngstown
East Elyria
Findlay
Mansfield
Wooster
Marion
Steubenville
Wheeling
Zanesville
Springfield
Dayton
Lancaster
Washington Court House
Marietta
Chillicothe
Athens
Portsmouth
Ashland
Huntington
Marketing Map of
OHIO
LEGEND
Principal Trading Center
Urban Cities 1,000 to 10,000 population
Less than 1,000 population
Trading Area Boundary
State Boundary

PLAINTIFF'S EXHIBIT 31

DATE: Feb 12 - 1964

AGREEMENT

AGREEMENT MADE by and between PERMA RESEARCH AND DEVELOPMENT COMPANY, a Delaware Corporation, with its principal place of business in North Attleboro, Commonwealth of Massachusetts, hereinafter referred to as "PERMA" and Richard H Travers and John H. Travers — 1143 Hancock St. Quincy, Mass.

PERMA does hereby appoint The above mentioned individuals and all other Perma Dealers as the Distributor of its "Perma Anti-Skid Control" and the Distributor accepts such appointment upon the following terms and conditions:

1. The area and territory for said distributorship shall be New England — (Except as see portion of Connecticut from Stamford)

as outlined in the unshadowed portion of the marketing maps attached to this agreement.

2. The Distributor shall employ the necessary sales personnel and service people to effectively merchandise the "Perma Anti-Skid Control" for said areas and territories which are the subject of said distributorship, as set forth in Paragraph 1.

3. The Distributor shall keep on hand and maintain a reasonable inventory of "Perma Anti-Skid Controls" and all parts incidental and necessary for the sale, repair, maintenance and installation of the same. After the expiration of six (6) months from the date hereof, a reasonable inventory shall be deemed to mean a thirty (30) day supply based on past experience.

4. The Distributor shall at all times employ a qualified mechanic to properly service and train other service and installation men. Said qualified mechanic shall be trained by PERMA at PERMA'S main office in NORTH ATTLEBORO, MASSACHUSETTS, at the sole cost and expense of the Distributor, including travel, maintenance and salary of its employees or personnel. PERMA, however, shall make no training charge thereof.

5. The Distributor shall conform to standards and technical instructions from time to time issued by Perma with respect to installa-

tion, service, maintenance, repair and replacement of "Perma Anti-Skid Controls" which are subject to this agreement, and to only use repair or replacement parts for such units which have been supplied or approved in writing by PERMA.

6. The Distributor shall, monthly and on such other times as PERMA may reasonably request, furnish PERMA reports of sales, inventory on hand and any other pertinent information on forms supplied by PERMA.

7. It is the intent of this Agreement that no products supplied by PERMA to the Distributor shall be sold, distributed, marketed or otherwise disposed of by the Distributor outside of the area and territory designated in Paragraph 1, and the Distributor shall use every effort to accomplish said intent and will take all necessary action on its part to enforce this intent in its dealings with its agents, dealers and customers in general.

8. The Distributor shall use its best efforts through its organization to distribute and procure sales, and create a demand for said "Perma Anti-Skid Control".

9. This Agreement is personal between PERMA and the Distributor, and shall not be assignable without the prior written consent of PERMA.

10. During the term of this Agreement or any extension thereof, the Distributor shall not market or otherwise deal with any other product or products which are competitive to the "Perma Anti-Skid Control".

11. The Distributor acknowledges that certain phases of the manufacture of the "Perma Anti-Skid Control" are outside the control of PERMA, and PERMA shall not be responsible for any delays in shipping said product for reasons beyond its control, including but not limited to strikes, fire, flood or other acts of God. PERMA, however, shall keep the Distributor periodically advised of the availability of the product for shipment.

12. The Distributorship herein granted shall be exclusive in the area and territory set forth in Paragraph 1, provided however, that PERMA reserves the right to make direct sales to national fleet accounts, original equipment manufacturers (such as automobile, truck and power vehicle manufacturers), manufacturers who supply components to the Original Equipment Manufacturers Market, U. S. Government Agencies, and national chain stores. National Chain Stores as herein used shall be defined as a

chain having ten or more retail outlets. In the event of any direct sales by PERMA to a National Chain Store, for re-distribution to branch stores, PERMA will ask for and use every reasonable effort to obtain the National Chain Stores Main Offices' co-operation in determining the units shipped to their branches as outlined in the territory described in Paragraph 1. Upon receipt of this information, PERMA shall pay the distributor 5% of the net billing price for those units that were trans-shipped into his territory. Said 5% re-imbursement shall be considered voluntary on the part of PERMA, and Distributor agrees that PERMA shall not be held liable if it is impossible or impracticable to obtain the necessary paper work from the National Chain Store, to issue said 5% over-ride.

13. PERMA shall supply literature and other sales data, which it may from time to time issue, to the Distributor on a cost plus handling basis, and shall furnish the Distributor with such other information as, in the opinion of PERMA, may be useful to the Distributor in the sale of "Perma Anti-Skid Controls".

14. PERMA shall refer to the Distributor all directly received inquiries and orders for "Perma Anti-Skid Controls", involving sales within the area and territory set forth in Paragraph 1 of this Agreement, with those exceptions as outlined in Paragraph 12 supra.

15. The Distributor shall accept separate billing of $1.85 each, not for each complete "Perma Anti-Skid Control" shipped to it; and PERMA shall contribute an equal amount each. Said total amount of $3.70 for each complete "Perma Anti-Skid Control" shall be set up as an advertising fund to be spent by the Distributor in accordance with PERMA'S advertising policy.

16. The term of this Distributorship shall be for the period from the date of this Agreement to [illegible]. The Distributor shall have an option to renew this Agreement for 5 successive annual terms, provided that (a) it shall have kept and performed all the terms and conditions of this Agreement, to be kept and performed by the Distributor during the preceding term; and (b) in the preceding term shall have ordered a minimum of Twenty-seven thousand (27,000) "Perma Anti-Skid Controls" or the average "Perma Anti-Skid Controls" per 1,000

of registered automobiles purchased by all other Distributors in their respective marketing areas whichever is greater. Average as herein used shall mean the total sales by all PERMA Distributors divided by the total number of automobiles registered in those Distributors' areas. Said options shall be exercised by United States Registered Mail addressed to PERMA at least sixty (60) days before the expiration of the prior term.

17. In case of failure on the part of the Distributor to pay for all merchandise sold to it by PERMA promptly when the same becomes due and payable, or in case the Distributor shall neglect or fail to perform or observe any of the terms and conditions imposed upon the Distributor by this Agreement, or in the event that the Distributor makes an assignment for the benefit of Creditors or is adjudged a bankrupt or a debtor, re-organization, arrangement, or similar petition or proceedings be filed by or against it under any chapter or provision of the Federal Bankruptcy Act so-called, or in the event a Receiver is appointed over the assets of said Distributor and said receiver is not vacated within ten (10) days thereafter; then and in any of the above events, PERMA shall at its option, and not withstanding any waiver of any prior breach of any covenant, term or condition, terminate this Agreement upon written notice to that effect addressed to the Distributor by United States Registered Mail, provided however, if the breach of any term or condition of this Agreement is of such nature that it may be cured or remedied, the Distributor shall have a period of thirty (30) days from the receipt of notice of said breach from PERMA to cure or remedy said default.

18. In the event of the termination of this Agreement for any cause whatsoever, PERMA shall have the right to purchase all its products in the possession of the Distributor at the then prevailing price therefor, minus a 15% handling charge.

19. (a) The Distributor does hereby order on the date hereof in proper form from PERMA a total of 27,000 Twenty seven thousand Complete "Perma Anti-Skid Controls" for the period starting with March 17 1964 and ending on March 17, 1965 The delivery schedule shall be as follows:

500 - March 1964 — 2500 - July - 1964 — 3500 - Nov 1964
500 - April 1964 — 2500 - August 1964 — 3500 - Dec 1964
500 - May 1964 — 2500 - Sept - 1964 — 3500 - Jan 1965
500 - June 1964 — 3500 - Oct - 1964 — 3500 - Feb 1965

(b) The cost to the Distributor for the complete "Perma Anti-Skid Control" shall be For /-six dollars and 27/100 ($46.27) plus three dollars and 70/100 ($3.70) Federal Excise Tax for each "Perma Anti-Skid Control" wit' freight F.O.B. North Attleboro, or Worcester, Massachusetts, U.S.A., it being specifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by PERMA to the Distributor. In order that the Distributor may have a flexible inventory we shall package Static Kits Variable Kits, and Adapter Kits separately.

1-Complete "Perma Anti-Skid Control" consists of:

1 - each Static Kit, Variable Kit, Adapter Kit.

(c) The variable kits and/or adapter kits portion is packaged separately and of necessity must be custom made to fit each automobile. Distributors may purchase additional quantities of variable kits and/or adapter kits as a separate item in order that their inventory of "Perma Anti-Skid Controls" be flexible at all times. Perma Research and Development Company shall exchange Variable Kits and/or Adapter Kits (in original packages) for Distributors (every 3 months) to keep their inventory under control and up to date at all times. There will be no charge for this exchange policy. The only expense to the Distributor will be transportation charges both ways.

(d) Payments to PERMA by the Distributor for Product shall be as follows: An escrow agreement shall be arranged between (Manufacturing National Bank) and Norfolk County Trust Co. of No Attleboro, Mass East Milton Branch. Milton, Mass for the first shipment (March) for subsequent shipments for April 1964 and May 1964 shall be on a C.O.D. basis. [illegible] 2% 10 days Subsequent shipments shall be on regular terms or 1% 10-30 days net

IN WITNESS WHEREOF, PERMA RESEARCH AND DEVELOPMENT COMPANY has caused these presents to be signed and its corporate seal to be hereunto affixed by Joseph Krawczyk, its Vice President and Treasurer.

[PX 31, p 6]

purpose duly authorized, and The Board of Directors has caused these presents to be signed and its corporate seal to be hereunto affixed by Clinton T. Crolius, its Secretary for this purpose duly authorized, this 24th day of February, A. D. 1964.

PERMA RESEARCH AND DEVELOPMENT COMPANY

By: [signature]

[signature], V Pres.

By: [signature]

[signature]

PLAINTIFF'S EXHIBIT 32 [PX 32, p 1]

A G R E E M E N T

Agreement between PERMA RESEARCH AND DEVELOPMENT COMPANY and MR. ANGELO DI SPIRITO, JR. and MR. CHRISTIE VITOLO, hereinafter referred to as the Distributor, for the Distributorship of the "Perma Anti-Skid Control".

1. The Distributor agrees to employ the necessary sales personnel and service people to effectively merchandise the "Perma Anti-Skid Control" for the areas and territories as follows:

Long Island - including the Boroughs of Brooklyn and Queens, in New York State.

2. The Distributor shall keep on hand and maintain a reasonable inventory of "Perma Anti-Skid Controls" and all parts incidental and necessary for the sale, repair, maintenance and installation of same.

3. The Distributor agrees at all times to employ a qualified mechanic to properly service and train other service and installation men. Said qualified Mechanic shall be trained by Perma at Perma's Main Office in North Attleboro, Massachusetts. Travel, maintenance and salary of said mechanic shall be borne by the Distributor.

4. The Distributor agrees to conform to standards and technical instructions from time to time issued by Perma with respect to sales, advertising, promotion, installation, service, maintenance, repair and replacement of Perma Anti-Skid Controls which are subject to this agreement, and to only use repair or replacement parts for such units which have been supplied or approved in writing by Perma, as the same shall apply to all distributors.

5. The Distributor shall keep Perma advised periodically, on forms to be supplied by Perma, as to the Distributors activities. The Distributor further agrees to answer all inquiries pertinent to promotion, maintenance, sales and development.

6. The Distributor warrants and guarantees that all product received from Perma will be sold, distributed or otherwise marketed only in the areas as designated in Paragraph 1. of this agreement and will use every effort legal or otherwise to accomplish the intent of this section which is that no product supplied by Perma shall be sold, distributed, marketed or otherwise disposed of outside of those areas as designated in Paragraph 1. and that it will take all necessary action on its part to enforce this intent in its dealings with its agents, distributors and customers in general.

7. The Distributor agrees that no arrangements or agreements shall be made with any of its agents, distributors, or customers inconsistent with this agreement.

8. The Distributor will ship and invoice at its own expense, and will attend credit risks of its own customers.

E-366

(paragraphs 7 and 8 transcribed on page E-366A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last two paragraphs thereof read as follows:

> "7. The Distributor agrees that no arrangements shall be made with any of its agents, distributors, or customers inconsistent with this agreement.
>
> "8. The Distributor will ship and invoice at its own expense, and will assume credit risks of its own customers."

9. The Distributor may assign the rights and duties under the contract to a Corporation managed by them within 30 days of the execution of this agreement. The Distributor shall not thereafter have the right to assign this agreement or Distributorship to any person, firm or Corporation without the written consent of Perma, but said consent shall not be unreasonably withheld.

10. The Distributor agrees that during the continuance of this agreement it will not market or otherwise deal with any other product or products which conflict with the marketing of the "Perma Anti-Skid Control" by the Distributor as provided herein.

11. Perma and the Distributor agree that best efforts shall be expended to comply with the terms of this agreement excepting that it is hereby acknowledged by the Distributor that certain phases of the manufacture of the "Perma Anti-Skid Control", is outside the control of Perma and Perma shall not be held accountable for delays in shipping product for reasons beyond its control which shall include but not be limited thereto, strikes, fire, flood or other acts of God, however in any event Perma shall keep the Distributor advised periodically of the availability of product for shipment.

12. It is hereby agreed by parties hereto that title to any product subject to this agreement shall remain in Perma until paid for by the Distributor as set forth herein and under this agreement. No product may be returned without prior written approval of Perma and Perma reserves the right to accept in original cartons free from damage, product returns at a 15% handling charge.

13. In the event that either of the parties hereto breach any of the conditions of this agreement, in any material manner, and said material breach is not cured within 30 days after notification in writing by registered mail return receipt requested, said notifying parties shall have the option to terminate this agreement without further notice.

14. (a) Perma agrees that the Distributor shall be the exclusive Distributor of the "Perma Anti-Skid Control" for the areas listed in Paragraph 1. of this agreement, however it is specifically agreed that Perma reserves the right to make sales direct to National Fleet Accounts, Original Equipment Manufacturers, such as automobile, truck and power vehicle manufacturers, manufacturers who supply components to the Original Equipment Manufacturer Market, United States Government Agencies and National Chain Store accounts, (See exception below) however, Perma shall use reasonable effort to refer to the Distributor all purchases of "Perma Anti-Skid Controls" by the foregoing accounts for consumption or sale in the areas designated in Paragraph 1. of this agreement, other than Original Equipment Manufacturers and Original Equipment Manufacturers Component Manufacturers, however it is specifically agreed that in no event shall Perma be held accountable or denied the right or be deemed to have relinquished any rights for sale or to sell any of its products to the above type of accounts and it is further specifically agreed that the Distributor shall not under any circumstances sell or arrange to sell any of the above type accounts, whose principal offices or place of business is located outside the designated areas in Paragraph 1. of this agreement.

(paragraph 14 of PX 32, p 2 is transcribed on page E-367A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last paragraph thereof reads as follows:

> "14. (a) Perma agrees that the Distributor shall be the exclusive Distributor of the "Perma Anti-Skid Control" for the areas listed in Paragraph 1, of this agreement, however it is specifically agreed that Perma reserves the right to make sales direct to National Fleet Accounts, Original Equipment Manufacturers, such as automobile, truck and power vehicle manufacturers, manufacturers who supply components to the Original Equipment Manufacturer Market, United States Government Agencies and National Chain Store accounts, (See exception below) however, Perma shall use reasonable effort to refer to the Distributor all purchases of 'Perma Anti-Skid Controls' by the foregoing accounts for consumption or sale in the areas designated in Paragraph 1, of this agreement, other than Original Equipment Manufacturers and Original Equipment Manufacturers Component Manufacturers, however it is specifically agreed that in no event shall Perma be held accountable or denied the right or be deemed to have relinquished any rights for sale or to sell any of its products to the above type of accounts and it is further specifically agreed that the Distributor shall not under any circumstances sell or arrange to sell any of the above type accounts, whose principal offices or places of business is located (?) the designated areas in Paragraph 1 of this agreement."

EXCEPTION

The exception to paragraph 14. (a) shall be that in the instance of National Chain Store Accounts, Perma Research and Development Company will negotiate and establish the price and policy with the National Chain Store Account for the purchase of the "Perma Anti-Skid Control" for the shipment and billing, by Perma Research and Development Company Distributors and Perma Research and Development Company will notify all Distributors of the Agreement and price schedules, which price shall be as close to the recommended Dealer Price List as possible, however, if Perma Research and Development Company must negociate on a Perma Research and Development Company to National Chain Store Account direct basis, it is understood that the Distributor will be reimbursed at a minimum rate of 5% of the National Chain Store Accounts billed price (excluding freight and F.E.T.) in the form of a credit to the Distributor.

15. (a) Perma Agrees to supply literature and other sales data, which it may from time to time issue, to the Distributor on a cost plus handling basis, and to furnish the Distributor with such other information as, in the opinion of Perma, may be useful to the Distributor in the sale of "Perma Anti-Skid Controls".

16. (a) Perma agrees to refer to the Distributor all directly received inquiries and orders for "Perma Anti-Skid Controls", involving sales within the areas designated in Paragraph 1. of this agreement, with the exception of O.E.M. Manufacturers and Component Part Manufacturers.

17. (a) It is hereby agreed that the Distributor hereby orders on the date hereof in proper form from Perma a total of Thirty-six thousand (36,000) "Perma Anti-Skid Controls" for the period starting with January 1, 1964 and ending on December 31, 1965. The delivery schedule shall be as follows:

A token quantity of 500-1000 units will be scheduled for delivery on November 15, 1963 or as soon as possible. The remaining units will be scheduled for delivery as follows:

2000	Units	- January, 1964
3000	"	- February, "
3000	"	- March, "
3000	"	- April, "
3000	"	- May, "
3000	"	- June, "
3000	"	- July, "
3000	"	- August, "
3000	"	- September, "
3000	"	- October, "
3000	"	- November, "
3000	"	- December, "

(b) The cost to the Distributor for the complete "Perma Anti-Skid Control" shall be Forty-six dollars and 27/100ths ($46.27) plus three dollars and 70/100ths ($3.70) Federal Excise Tax for each "Perma Anti-Skid Control" with freight F.O.B. North Attleboro, or Worcester, Massachusetts, U.S.A. It being specifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by Perma to the Distributor. In order that the Distributor may have a flexible inventory we shall supply Static Kits, Variable Kits, and Adapter Kits on the following cost price basis:

[PX 32, p 3]

(the last paragraph of PX 32, p 3 is transcribed on page E-368A)

After viewing all available copies of the preceding page, counsel for Appellant believes that the last paragraph thereof reads as follows:

> "The cost to the Distributor for the complete 'Perma Anti-Skid' (?) shall be Forty-six dollars and 27/100ths ($46.27) plus three dollars and 7(?)/100 ths ($3.7?) Federal Excise Tax for each 'Perma Anti-Skid Control' with (?)... North Atteboro, or Worcester, Massachusetts, U.S.A., it being specifically agreed that said price for said units under the same terms shall be subject to change by giving at least seven (7) days prior notice in writing by Perma to the Distributor. In order that the Distributor may have a flexible inventory we shall supply Static Kits, Variable Kits, and Adapter Kits on the following cost price basis:"

Static Kit---(Universal for all cars)	$70.30	$32.54	$2.61	$35.15
(Custom designed fittings				
Variable Kit-for make and model car)	21.85	10.58	.84	11.42
Adapter Kit--(Speedometer & Anti-Skid drive adapter)	7.80	3.15	.25	3.40
TOTALS	$99.95	$46.27	$3.70	$49.97

(c) The variable kits and/or adapter kits portion is packaged separately and of necessity must be custom made to fit each automobile. Distributors may purchase additional quantities of variable kits and/or adapter kits as a separate item in order that their inventory of "Perma Anti-Skid Controls" be flexible at all times. Perma Research and Development Company agrees to exchange variable kits and/or adapter kits for Distributors (every 3 months) to keep their inventory under control and up to date at all times. There will be no charge for this exchange policy. The only expense to the Distributor will be transportation charges both ways.

(d) Payments to Perma by the Distributor for Product shall be as follows:

November, December, January, February and March shipments will be on a payment in advance basis less 1% cash discount.
Starting with April Shipment the terms shall be, 1% cash discount ten (10) days from date of invoice, or 30 days net, and continue for the balance of this agreement, providing that the Distributors financial statement and balance sheet shall show a cash balance equal to the amount that is scheduled to be shipped by Perma, as shown on the above schedule.

18. The Distributor agrees to accept separate billing of one dollar and 05/100ths ($1.05) each, net for each complete Perma Anti-Skid Control shipped.

PERMA RESEARCH AND DEVELOPMENT COMPANY agrees to contribute an equal amount of one dollar and 05/100ths ($1.05) each for the "Perma Anti-Skid Controls" shipped to the Distributors.

The above totals of Three Dollars and 70/100ths ($3.70) shall be set up as an accumulative advertising fund to be spent by the Distributors in accordance with Perma's advertising policy. (See attachment #1)

19. The Distributor shall have the option to renew this agreement on or before December 15, 1964 and each November 15th thereafter until on or before November 15, 1969 provided that on the prior August (12) months the Distributor shall have purchased a minimum of Thirty-six thousand (36,000) [illegible] or the average "Perma Anti-Skid Controls" per [illegible] purchased by all other distributors in [illegible] Distributor shall not then be in material default in [illegible] agreement.

[illegible] of additional purchases as outlined in 6. (a) is as follows:

[illegible] sales by all Perma Distributors [illegible] by total number [illegible] Distributors [illegible]

After viewing all available copies of the preceding page, counsel for Appellant believes that the last four paragraphs thereof reads as follows:

"The above totals of Three Dollars and 70/100 ths ($3.70) shall be set up as an accumulative advertising fund to be spent by the Distributor in accordance with Perma's advertising policy. (See attachment 1).

19. The Distributor shall have the option to renew this agreement on or before November 15, 1964 and each November (15th) thereafter until November 15, (1974?) provided that in the prior twelve (12) months the Distributor shall have received a minimum of Thirty-six thousand (36,000) (?)... or the average 'Perma Anti-Skid Controls' (?)... (preceeded?) by all other distributions in (?)... (is greater and?) (provided?) (?) further that the (?)... shall not then be in (?)... under (?)... this agreement." (Illegible to the end of the page)

to be maintained.

(a) Advance notice of 60 days and not more than 120 days by registered mail, return receipt requested to Perma's usual place of business is required for renewal. This agreement shall expire on November 15th in any of the years 1964 thru 1973 if not theretofore renewed in such year as above provided, and shall expire in any event on November 15, 1974.

20. Perma hereby grants to the Distributor the following options:

(a) The exclusive right to distribute in the territory described in Section #1 all future Perma products.

(b) To be Perma's exclusive distributor in the territory described in exhibit "B" attached hereto said area being described as the New York City Marketing Area.

Provided:

1.- Said option is exercised by notice in writing on or before July 1, 1964.
2.- Distributor purchases, in addition the number of units described in 17.(a) above, 64,000 units per year in equal monthly shipments, beginning the 12 months, ending June 30, 1965, all other terms and conditions of this agreement to apply equally to said New York City Marketing Area territory.
3.--In granting this option Perma makes no present commitment to deliver such additional units but will use its best efforts so to do.

In presence of:

PERMA RESEARCH AND DEVELOPMENT COMPANY

BY: Sam Karp
Sam Karp, Vice President Marketing

BY: _____
Title

BY: _____
Title

NO MATERIAL ON PAGE E-371

[PX 34, p 1]

Recalled

PLAINTIFF'S EXHIBIT 34

This Agreement made this 1st day of January, 1965 by and

BETWEEN: SINGER COMPANY OF CANADA LTD.

a Quebec Corporation, having its registered office at 1245 Sherbrooke Street West, Montreal, (hereinafter called "SINGER-CANADA")

AND O'MATIC AUTO PRODUCTS LTD.

a corporation having an office at 407 Fina Building, Calgary, Alberta, (hereinafter called O'MATIC").

WITNESSETH:

WHEREAS SINGER-CANADA is prepared to grant exclusive selling and distribution rights to O'MATIC subject to the conditions hereinafter set forth, for a certain invention known as "Perma Anti-Skid Control" (hereinafter referred to as the "Product"), including any parts, improvements or modifications incorporated therein, during the term of this Agreement and any extensions or renewal thereof;

NOW THEREFORE, in consideration of the premises and the mutual promises herein contained and for other good and valuable consideration, it is agreed as follows:

1. GRANT

SINGER-CANADA hereby grants to O'MATIC upon the terms and conditions and with the limitations hereinafter set forth, exclusive rights to market, sell and otherwise merchandise and promote the Product in Canada, provided, however, that SINGER-CANADA does not grant under this Agreement the rights to promote, distribute and sell the Product to original equipment manufacturers. The rights granted to O'MATIC hereunder are limited solely to Canada and O'MATIC shall have no rights in connection with the Product whatsoever outside of Canada.

2. PATENTS, TRADEMARKS AND COPYRIGHTS

O'MATIC agrees that all patent trademarks and copyrights now or hereafter obtained by SINGER-CANADA, or its heirs, successors or assigns in any jurisdiction in respect of the Product, or any part or process

...relating

△#385

[PX 34, p 2]

relating thereto, shall be the exclusive property of SINGER-CANADA. O'MATIC will not dispute the validity of any such patents, trademarks or copyrights or SINGER-CANADA's title thereto and will not aid others in doing so and will not avoid or attempt to avoid its obligations hereunder because of any asserted invalidity or contested ownership of SINGER-CANADA thereto.

3. PROMOTION

O'MATIC hereby accepts upon the terms and conditions of this Agreement the rights herein granted and agrees to continuously sell, merchandise and promote throughout Canada the sale of, and to use its best efforts to increase the demand for the Product.

Upon termination or expiration of this Agreement as herein provided, O'MATIC agrees that it will not for a period of ten years thereafter engage in the manufacture, sale or distribution of ANTI-SKID devices of any sort and will not have any interest in any company or concern so engaged.

4. MASTER SAMPLE

SINGER-CANADA hereby agrees that all Product sold to O'MATIC by SINGER-CANADA shall be manufactured in a good and workmanlike manner and in accordance with specifications which shall match in performance a master sample previously tested and approved by O'MATIC and agrees to provide a marketable product.

5. CLAIMS

Any claims by O'MATIC on SINGER-CANADA for returns, damages or expenses occasioned by the failure of Product to operate due to poor workmanship, poor quality or faulty design shall be limited to the purchase price of the product as covered by O'MATIC Purchase Orders to SINGER-CANADA.

SINGER-CANADA shall indemnify and hold O'MATIC harmless from any damages, costs and penalties incurred by O'MATIC as a result of any claim for injury to a user or users of the Product arising out of or due

[PX 34, p 3]

due to faulty workmanship in the manufacture and assembly of the Product or for any fault in the design.

O'MATIC shall indemnify and hold SINGER-CANADA harmless from any damages, costs and penalties incurred by SINGER-CANADA as a result of any claim for injury by a user or users of the Product, arising out of or due to improper or faulty installations or applications of the Product.

6. INFRINGEMENTS

In the event that O'MATIC shall learn of an infringement of any of SINGER-CANADA's patents relating to the Product, O'MATIC shall notify SINGER-CANADA thereof in writing, and provide SINGER-CANADA with any available evidence of infringement. In any infringement suit which SINGER-CANADA may institute, O'MATIC shall, at the request and expense of SINGER-CANADA, co-operate with SINGER-CANADA in all respects, making available to SINGER-CANADA all relevant records, papers, information, samples, specimens and the like and make all reasonable efforts in good faith to secure the testimony of its employees and other evidence available to it.

7. PATENT NOTICE

All Product sold by O'MATIC under this Agreement will bear a patent notice approved by SINGER-CANADA in compliance with applicable statutory requirements. O'MATIC will incorporate in all advertisements, press releases or promotional literature of any type the name "Perma Anti-Skid Control" or such other trade name as SINGER-CANADA may establish from time to time.

8. ADVERTISING

O'MATIC agrees that all its advertising, press releases, and other promotional literature and activities relating to the Product shall be submitted to SINGER-CANADA in advance of the use thereof and O'MATIC agrees not to use any such literature or activities if SINGER-CANADA shall reasonably object thereto.

O'MATIC agrees to identify, offer, sell, advertise and promote the Product only by such names and trademarks as may, from time to time, be authorized

(paragraphs 7 and 8 transcribed on page E-366A)

authorized in writing by SINGER-CANADA.

9. PURCHASE ORDER

O'MATIC hereby agrees to purchase Product from SINGER-CANADA, in accordance with the purchase order attached hereto and marked EXHIBIT A.

10. AUTOMATIC TERMINATION

In the event of failure on the part of O'MATIC to pay any sums due to SINGER-CANADA for Product or to perform any other of its obligations hereunder and such failure shall continue for a period of 30 days after notice in writing thereof from SINGER-CANADA, or in the event that O'MATIC either voluntarily or involuntarily makes an assignment for the benefit of creditors or becomes a bankrupt or becomes a party to any winding up of its affairs, this Agreement and all rights granted O'MATIC hereby or hereunder shall immediately be subject to termination by SINGER-CANADA at its sole option.

11. ASSIGNMENT

Neither this Agreement nor any rights or part thereof granted hereby or hereunder shall be granted, conveyed or assigned by O'MATIC without the written consent of SINGER-CANADA.

12. NOT AN AGENT

Nothing herein contained shall be construed to constitute O'MATIC an agent of SINGER-CANADA for any purpose.

13. TERM

This Agreement shall become effective and commence as of the day and year first above written, and shall continue in force and effect for twelve (12) months. Thereafter, in each year on or before the anniversary of this Agreement for a period of nine (9) successive years, O'MATIC shall have an option to renew for a further period of twelve (12) months provided however

a) that O'MATIC shall have placed Purchase Orders with SINGER-CANADA for delivery of a minimum of twelve thousand (12,000) units of

....Product

Product during the twelve (12) month period then ended; and provided

b) that O'MATIC shall have carried out all its duties and met all its obligations under this Agreement; and provided

c) that both parties agree, in writing, on any price adjustments, delivery schedules, quantities and any other terms and conditions to be applicable during such additional twelve (12) month period.

14. CESSATION OF RIGHTS

Upon termination of this Agreement, all of the licenses, rights and privileges granted hereunder by SINGER-CANADA to O'MATIC shall forthwith terminate and cease.

15. FORCE MAJEURE

SINGER-CANADA shall not be liable for any loss, damage, or expense of O'MATIC or any other person arising from any delay or failure by SINGER-CANADA to make deliveries in accordance with EXHIBIT A, or any other delivery schedule for the Product hereunder, owing to force majeure, strikes, riots, civil disturbance, insurrections, restraint of princes, acts of Government, or any other circumstances beyond the control of SINGER-CANADA.

16. NOTICE

Any notice or communication under this Agreement shall be deemed to have been duly given if mailed by registered or certified mail postage prepaid to the other party at its place of business hereinabove mentioned, or at such other address as may be furnished to the notifying party in writing.

17. ENTIRE AGREEMENT

This Agreement constitutes the entire agreement between the parties hereto. This Agreement may not be changed or modified except in writing.

18. WAIVER OF RIGHTS

No failure by either party to take action in respect of any default or breach hereunder shall be deemed to be a waiver of any provisions of this Agreement and no waiver of any default or breach or failure to take action in respect thereof shall be deemed to extend

to

(paragraph 14 of PX 32, p 2 is transcribed on page E-367A)

[PX 34, p 6]

to any other default or breach.

IN WITNESS WHEREOF, the parties hereto have caused this Agreement to be executed as of the day and year first above written.

SINGER COMPANY OF CANADA LTD.

By _____

ATTEST:

_____ SECRETARY

O'MATIC AUTO PRODUCTS LTD.

By _____

ATTEST:

_____

APPROVED BY
THE SINGER COMPANY

By _____

ATTEST:

_____

[PX 35, p 1]

PLAINTIFF'S EXHIBIT 35

"A"

This AGREEMENT made this 18th day of June, 1964, by and between PERMA RESEARCH AND DEVELOPMENT COMPANY, a Delaware corporation, having its principal place of business at 345 East Washington Street, North Attleboro, Massachusetts (hereinafter called "Perma"), and THE SINGER COMPANY, a New Jersey corporation, having a place of business at 321 First Street, Elizabeth, New Jersey (hereinafter called "Singer"),

WITNESSETH:

WHEREAS, Perma wishes to establish a manufacturing source for an apparatus known as the "Perma Anti-Skid Control", a device consisting of a so-called Static Kit, Variable Kit and Adapter, and related parts and components, substantially in accordance with plans and specifications heretofore delivered by Perma to Singer with such modifications therein as have been heretofore disclosed by Perma to Singer (hereinafter, including any improvement or modification incorporated therein during the term of this Agreement and any extension thereof, called the "Product"), and

WHEREAS, Singer is willing to manufacture the Product for Perma and to assume responsibility for such manufacture in the United States upon the terms of this Agreement;

NOW, THEREFORE, in consideration of the mutual covenants, conditions and agreements hereinafter contained, it is hereby agreed as follows:

1. PATENTS

Perma represents that it is the owner of the entire right, title and interest in and to the following duly issued and existing United States patents, and the following pending applications for United States patents covering the Product (hereinafter collectively called the "Patents"):

Patent: 2,881,879 Issued Apr. 14, 1959

Applications: SN 203,075 Filed June 18, 1962
SN 322,716 Filed Nov. 1963

2. GRANTS

Perma hereby grants to Singer under the Patents the sole and exclusive right and license to make and/or assemble the Product in the United States and to sell the same to Perma and/or to customers designated by Perma. Perma reserves to itself the right to sell Products in the United States, provided said Products are made or assembled hereunder by Singer, and to make and/or assemble Products for experimental testing and developmental purposes, but not for sale. Singer shall have the right to sublicense others to make and/or assemble the Product and parts and components thereof.

3. SECOND SOURCE

Should Perma and/or Singer deem it advisable to establish a second source for the Products in the United States, Singer shall, if it chooses to do so, have the responsibility and the right to locate and sublicense a manufacturer for such purpose. Singer shall notify Perma within thirty (30) days after a request from Perma to do so, whether it will undertake to establish

such a source and shall keep Perma advised of its efforts in doing so. If Singer declines to do so, then Perma shall have the right to locate and license a manufacturer for such purpose, which shall not otherwise affect the rights and licenses granted to Singer hereunder.

4. SUPPLIERS

Singer has the right and assumes the responsibility to deal with vendors and suppliers of parts, components, or assemblies of the Product, including the right to negotiate price, delivery, quantity, quality, and other terms and conditions under which materials and supplies will be furnished, in order to provide Perma with completed Products. Singer may at any time change vendors and suppliers and shall give Perma notice of such changes from time to time.

5. PURCHASE ORDERS

Perma shall issue purchase orders to Singer, the first of which shall be for 50,000 Products and shall be issued simultaneously with the taking effect of this Agreement. Finished Products shall be shipped by Singer from time to time only in accordance with Perma's written instructions, provided that Singer shall not be liable for damages under this Agreement for failure or delay in performance of the provisions hereof caused by strikes, other labor difficulties, wars, fires, floods, explosions, government regulations, court orders, or other events beyond its control. If Singer is unable to meet such schedules because of any of the foregoing

events, Singer shall use its best efforts to fill Perma's purchase orders in accordance with this Agreement through substitute means.

Any dispute as to failure to comply with the provisions of this paragraph 5, and any claim as to damages resulting therefrom, shall be the subject of arbitration in accordance with the provisions of paragraph 15.

6. FORECAST OF REQUIREMENTS

On the first business day of each month during the term hereof, Perma shall supply Singer with a written estimate of its monthly requirements of the Products during the three-month period commencing thirty (30) days from the date of such estimate. Such estimates shall be subject to revision in subsequent monthly estimates.

7. PRICE

Perma shall pay Singer for each Product shipped by Singer a price stated in the purchase order relating thereto, which price shall be in accordance with Schedule A, attached hereto, for the first purchase order for 50,000 Products being issued herewith.

Prices for Products covered by subsequent purchase orders shall be as agreed upon between the parties hereto; upon failure of the parties to agree upon any such prices within thirty (30) days after the submission of a proposed purchase order by Perma, such prices shall be the subject of arbitration in accordance with the provisions of paragraph 15. It is the intention of the parties that changes in costs shall be equitably reflected in prices for Products in any such subsequent

[PX 35, p 5]

purchase order.

8. BILLING AND PAYMENT

On the first, tenth and 20th of each month, or on the first business day thereafter, Singer shall supply Perma with a statement showing the number of Products shipped, the prices at which the same were invoiced and the total amount payable to Singer therefor, including any shipping charges paid by Singer. Perma shall remit to Singer the total amount shown to be payable within thirty (30) days of the date of each such statement. Perma shall be given credit on a current basis for Products accepted by Singer for return.

Simultaneously with the shipment of Products, Singer shall invoice the customers upon forms to be supplied by Perma.

9. IMPROVEMENTS

Perma and Singer agree to assist each other in maintaining joint continuing engineering supervision and development of the Product, and agree to make available to each other their technical knowledge applicable to the design and manufacture of the Product. Perma and Singer shall each promptly disclose to the other any and all improvements in the Product and in the manner of making or assembling the same which each respectively shall develop or acquire, whether patentable or not, during the term of this Agreement, and Singer shall be free to make use of the same in making and/or assembling the Product hereunder. Any patent obtained thereon by Perma shall be deemed a Patent hereunder and Perma shall have a royalty free license to make, have made, use and sell under any patent obtained thereon by Singer for the

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full term of any such patent.

No change in design or materials of the Product shall be made by Singer unless and until the same shall have been approved in writing by at least two officers of Perma, including any change or deviation with respect to the parts or components acquired by Singer by operation of the agreement of even date herewith between Singer and [redacted] parts or components are determined to be useable by Frank Perrino, pursuant to paragraph 3 of the agreement among American Emblem Company, Inc., Worcester Stamped Metal Company and Singer of even date herewith, such determinations shall be binding upon Perma hereunder and shall constitute approval of a change hereunder. If any such approved change in design or materials should result in a change in the cost of manufacture, the price of the Product shall thereafter be as agreed upon between the parties. Upon failure of the parties to agree upon any such prices within thirty (30) days after the approval of such change in design or materials such price shall be the subject of arbitration in accordance with the provisions of paragraph 15.

10. WARRANTY AS TO QUALITY

Singer shall manufacture the Products in accordance with specifications and blueprints furnished by Perma and shall provide diligent quality control in the production, assembly, testing and packaging of the Products. Singer hereby warrants that all of the Products delivered hereunder shall be of first class materials and workmanship, except that Singer shall not be liable hereunder to Perma

(a) with respect to the materials, provided they comply with the specifications furnished or approved from time to time by Perma, and

(b) with respect to the parts or components acquired by Singer by operation of the agreement of even date herewith among Worcester Stamped Metal Company, American Emblem Company,

[PX 35, p 7]

Inc., and Singer if Singer notifies Perma that any of such parts and components are not in accordance with Perma's specifications and Perma grants a deviation therefrom for such parts and components.

11. TOOLING

Singer shall have complete freedom to move to manufacturing facilities of its choice any tooling presently owned by Perma which may be useful in the manufacture of the Products, for use during the term of this Agreement, provided, however, that such tooling shall remain the property of Perma. The cost of moving such tooling to such facilities, and the cost of altering the same, shall be borne by Singer. The cost of any new tooling required in the manufacture of the Product shall be borne by Singer, and such tooling shall belong to Singer.

Upon termination of this Agreement, Perma shall purchase said tooling from Singer at a price to be agreed upon between them based upon its remaining useful life. Upon failure of the parties to agree upon any of such prices within sixty (60) days after termination, said prices shall be the subject of arbitration in accordance with the provisions of paragraph 15. The cost of moving or altering such tooling following termination hereof shall be borne by Perma.

12. INDEMNIFICATION

A. Perma shall indemnify Singer against and shall hold Singer harmless from any and all damages, costs and penalties incurred by Singer as a result of

(a) infringement or alleged infringement by Singer of any patent owned or controlled by a person or corporation not a party to this Agreement arising out of the manufacture, use or sale of the

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Products; and

(b) any claim or suit for injury to the user of any Product arising out of or due to any fault in the design of the Product approved by Perma.

Singer shall notify Perma of any and all such claims or suits against Singer and Perma shall undertake and maintain the defense of any and all such claims and suits. Singer shall have the right to participate by counsel of its own choosing in defense of any and all such suits at Singer's expense.

D. Singer shall indemnify Perma against and shall hold Perma harmless from any damages, costs and penalties incurred by Perma as a result of any claim or suit for injury to the user of any Product arising out of or due to any defect in the materials or workmanship of the Product in so far as the same are warranted by Singer in Paragraph 10 above. Perma shall notify Singer of any and all such claims or suits against Perma and Singer shall undertake and maintain the defense of any and all such claims and suits. Perma shall have the right to participate by counsel of its own choosing in the defense of any and all such suits at Perma's expense.

C. The provisions of this Paragraph 12 shall survive any termination of this agreement with respect to any Product sold by Singer prior to such termination.

13. PROTECTION AGAINST THIRD PARTY INFRINGERS

Perma and Singer shall each give the other prompt notice of any infringement by any other of any Patent of which it has knowledge. Upon request of

[PX 35, p 9]

Singer, Perma shall protect Singer from infringement by any other of any and all patents to the extent of bringing suit against any infringer or infringers thereof to enjoin or otherwise to compel cessation of such infringement. In the event that Perma shall, when so requested by Singer, refuse to bring any such action, Singer may, at its option, but at Perma's expense and in Singer's own name or in the name of Perma, take such action as it may deem necessary to end such infringement.

14. TERM AND TERMINATION

A. Upon completion and shipment by Singer of 250,000 Products, or in June, 1966, whichever event shall first occur, the parties hereto shall confer for the purpose of determining whether the Agreement is to continue in effect either upon the same or upon other mutually acceptable terms and conditions. If the parties are unable to reach agreement within sixty (60) days of such event, either party shall have the right to give written notice to the other party of its desire to terminate this Agreement six (6) months following the giving of such notice. Unless and until such notice is given, this Agreement shall continue in full force and effect. If either party so elects to terminate, Singer shall continue to meet Perma's reasonable shipping requirements on a decelerating basis during the six-month period following notice and before termination. Upon termination, however caused, Singer shall assemble into complete units as far as possible all component parts for the Products in its possession and Perma shall purchase such completed Products and any remaining parts and components at the price therefor last in effect under this Agreement.

[PX 35, p 10]

D. In the event a material default by either party with respect to any provision of this agreement shall occur and such default shall continue for 30 days after written notice from the other party specifying such default, then the party giving such notice may terminate this agreement effective not less than 90 days subsequent to the giving of such notice. Notice of such termination shall be given at least sixty (60) days prior to the effective date of termination specified in such notice, and such notice having been given, this Agreement shall terminate on such date.

C. In the event the manufacture and sale of the Product under this agreement by Singer shall in the opinion of competent counsel satisfactory to the parties hereto constitute an infringement of the patent rights of any other person in any material respect or in the event that Perma shall be adjudicated a bankrupt or an order shall be entered making Perma the subject of any proceedings for reorganization, dissolution, liquidation, winding up or adjustment of its obligations or any class of its obligations, or if Perma shall make an assignment for the benefit of creditors or a receiver shall be appointed for its property or any substantial portion thereof, or if Perma shall voluntarily liquidate, then in any such event Singer shall have the right, upon the giving of written notice to Perma or its successor in interest, forthwith to terminate this Agreement.

D. In the event that Singer shall be adjudicated a bankrupt or an order shall be entered making Singer the subject of any proceedings for reorganization,

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dissolution, liquidation, winding up or adjustment of its obligations or any class of its obligations, or if Singer shall make an assignment for the benefit of creditors or a receiver shall be appointed for its property or any substantial portion thereof, or if Singer shall voluntarily liquidate then upon the happening of any such event this Agreement shall automatically, and without further action or notice, terminate.

15. ARBITRATION

Arbitration, in those instances in which the same is hereinabove provided for, shall be conducted by, under, and in accordance with the rules then obtaining of the American Arbitration Association. In the event of a dispute and upon the failure of the parties to have reached agreement within the period if any, specified in each such instance, either party may by written notice advise the other that it intends to submit such dispute to arbitration either in New York, New York or in Elizabeth, New Jersey, and the notifying party shall thereupon be free to institute such arbitration proceeding. The award rendered in such arbitration shall be final and conclusive and binding upon each party without any right to judicial or other review.

16. NOTICE

Any notice or communication under this Agreement shall be deemed to have been duly given if mailed by registered or certified mail, postage pre-paid, to the other party at its place of business herein above mentioned or at such other address as may be furnished to the notifying party in writing.

[PX 35, p 12]

17. APPLICABLE LAW

This Agreement shall be construed in accordance with the law of New Jersey.

18. ASSIGNABILITY

This Agreement shall be binding upon and shall inure to the benefit of the parties hereto and their successors and assignees. Perma shall not, however, sell or assign any of its rights in the Product and/or any of the Patents unless and until it shall have first offered to Singer the opportunity to purchase said rights upon the same terms and conditions, which offer shall have been open to Singer for a period of at least thirty (30) days.

IN WITNESS WHEREOF, the parties have executed this Agreement as of the date first above written.

THE SINGER COMPANY

ATTEST: By C. S. McCann

PERMA RESEARCH AND DEVELOPMENT COMPANY

ATTEST: By Leonard P. Ferrara Pres.

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[PX 35, p 13]

PERMA ANTI-SKID DEVICE

	Freight*	Price, Incl. Freight
Static Kit	$1.00	$30.00
Variable Kit	.10	3.50
Molded Adaptor	.02	1.00
Machined Adaptor	.05	2.60
TOTAL W/MOLDED ADAPTOR		34.50
TOTAL W/MACHINED ADAPTOR		36.10

*Following the delivery of the first 5,000 units the freight cost will be re-examined and, if necessary, an adjustment made to reflect these costs more accurately.

SCHEDULE A

NO MATERIAL ON PAGES E-391 through E-401

tained, it is hereby agreed as follows:

[PX 38A, p 1]

DEF. Ex. 38.

PLAINTIFF'S EXHIBIT 38A

Memo: Perma Anti-Skid Unit

Subject: Meeting between Perma and Singer Personnel

Date: September 2, 1964

In Attendance: Messrs.

F. Di Benedetto
R. Mainelli
H. Snyder, Jr.
A. Romel
R. R. Patten
J. F. Moore
E. Kinal
H. Sharp

This meeting was held to determine all problems which are preventing the production of satisfactory Anti-Skid Units.

The following items were discussed.

1. Engineering Data

Various part prints and material specifications of parts to be obtained from Perma or Vendors. In the event that part prints or specifications are not obtainable, Singer will measure and draw up parts and apply tolerances. Perma will then approve these prints and tolerances. Specifications on various assemblies will be written up by Singer and Perma will then approve them.

2. Testing

Perma will approve test specifications on the following:

Perma-vac Unit
Sensing Device
Pressure Switch

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[38A, p 1]

3. Defective Parts

Pressure Switch:

Excessive leaking through the pressure setting screw caused the examination of several disassembled switches. This disclosed that the Piston "O" Ring was being cut due to a sharp edge on the I. D. of the piston hole. This defect is very serious and could cause a brake failure. This defect was brought to the attention of U. S. Dielectric and they promised to correct this condition. Inspection of the "O" Rings which they are presently using have a heavy flash on the O. D. and I. D. In the future they agreed to use Conover "O" Rings which are of a better quality.

L. Mager of U. S. Dielectric brought down six pressure switches which were supposed to be set within 85-100 p.s.i. These switches were checked on our Test Stand and they showed approximately 20# out of specification. It was decided to have our gage calibrated and this was done the same day by U. S. Testing Company, Hoboken, New Jersey. The gage was reinstalled on the stand and is now accurate within ± 1 p.s.i. The switches from U. S. Dielectric were rechecked and showed 20# - 30# high. Since our gage before calibration read 5# low, this means that the gage on U. S. Dielectric's Test Stand must be 25# high. It may be necessary to take our master gage to U. S. Dielectric and check their gage calibration.

4. Sensing Units

Sixteen (16) Sensing Units were disassembled by Perma personnel and the timing of the microswitch was found to be off on two (2) units.

One unit had a bent gear shaft pin and a friction spring which was not centralized on the cam washer.

Another unit had a poor bearing on the cam gear thrust washer.

A loose wire due to faulty soldering was found and also there was no oil on the gear shaft.

[PX 38A, p 2]

Perma also requested that in the future two (2) drops of oil on the gear shaft and two (2) drops of oil be applied on the opposite side through the small filter.

American Optical was notified of the necessity of performing this operation. They also were advised of the type of oil to be used and the method of application.

It was agreed by Perma and Singer that a representative of American Optical would come down and disassemble fifteen (15) Sensing Units to verify defects found by Perma.

5. Gear Boxes

Perma informed us that after a Life Test was run on American Optical Gear Boxes, they find that they do not hold up as well as the Gear Boxes produced by Swank.

The prints of the Gear Boxes and also a comparison chart between American Optical tolerances and Swank tolerances have been given to Mr. E. Kinal of Quality Control on September 4th. Mr. J. Moore called American Optical and told them to halt production of Gear Boxes until further notice. Singer will make a dimensional check on Gear Boxes purchased from American Optical to determine if they meet print specifications. If the parts are to print and additional research and product testing is required to find a solution to this problem, Perma has agreed to work out a financial arrangement with Singer for this work.

H. Sharp

H. Sharp
Supervisory Engineer
Manufacturing Engineering
Contract Sales

HS/ms

cc: J. Fecko
Those attending

[PX 38A, p 4]

DEF. EX. 71.

Memo: Perma Anti-Skid Unit

Subject: Meeting between Perma, American Optical and Singer personnel

Date: September 3, 1964

In Attendance: Messrs.

F. Perrino	H. Snyder, Jr.
R. Mainelli	R. R. Patten
J. R. Vairo	A. Romel
R. Lee	H. Sharp
E. Kuzdzal	E. Kinal
N. Lapine	J. F. Moore
	R. Kloby

With the above personnel present, fifteen (15) Units were disassembled, and the following defects were found.

1. Gear Shaft not central (one (1) part very bad and two (2) units questionable. American Optical agreed that one (1) part was very bad and could cause excessive wear. This condition was brought to American Optical's attention previously by Perma.

2. Cam Button was not seated squarely on top of cam. This defect was found on one (1) unit. This is not a serious defect but should be looked into.

3. Friction Spring not aligned properly. This defect according to Mr. F. Perrino is serious and is cause for rejection. This condition was found by Mr. F. Perrino on one (1) unit.

4. Cam Thrust Washer Bearing did not have a good finish. This defect was found on one (1) unit.

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5. Governor spool face - two (2) parts had rough finish and one (1) part had peeling plating. Specification on drawing for this part calls for 16 R.M.S. and a ground surface on face. Mr. F. Perrino agreed that this surface could have a surface finish of 63 R.M.S. but the plating should not peel.

6. Setting on micro switch - two (2) parts had high setting of 3-1/4 to 3-3/8 instead of specification 2-3.

American Optical was notified to halt production of Sensing Units until the above problems were ironed out and an acceptable quality level was established.

Perma has a check list of twenty-five (25) characteristics which they feel are necessary to insure quality of Sensing Devices. Several more characteristics are being added to this list by Singer.

These items were given to American Optical by Perma verbally. Singer will formalize these checks as an assembly specification.

Mr. F. Perrino of Perma is particularly concerned about the Governor spool plating, the micro switch settings and proper positioning of the thrust washer; and that the solenoid meets all assembly specifications.

A suggestion was made by Mr. H. Snyder that the 15,000 Sensing Units on hand which require an extra gasket and washer be checked for micro switch setting and other elements under the cover. It will be necessary to obtain an electrical tester similar to American Optical's to check the micro switch settings. It was agreed that American Optical would work out a financial arrangement with Singer for this additional work.

American Optical did not feel that a decision could be made as to the extent of their liability to correct the above defects, or to what quality level to agree to until additional data was obtained and studied. Therefore, it was decided that American Optical and Perma would send representatives to Singer on Tuesday, September 8th, and one-hundred (100) units would be selected at random from several different shipments and checked. The defects found would be noted on a tag which would be attached to the defective unit. In this way, a frequency of similar or critical defects can be ascertained. These results will be given to each company for evaluation.

A meeting will be held on September 14th between Perma, American Optical and Singer to discuss the results and reach a final decision on the disposition of the Sensing Units.

H. Sharp

H. Sharp
Supervisory Engineer
Manufacturing Engineering
Contract Sales

HS/ms

cc: J. Fecko
Those attending

PERF. 40.

Memo: Perma Anti-Skid Unit Sensing Device

Subject: Meeting between Perma, Singer and American Optical personnel

Date: September 8, 1964

In Attendance: Messrs. F. DiBenedetto (Perma)
R. Mainelli (Perma)
N. Lapine, Sr. (American Optical)
R. E. Lee (American Optical)
E. Kinal (Singer)
J. W. Casner (Singer)
H. Sharp (Singer)

This meeting was held to determine the quality level present in the Anti-Skid Unit Sensing Device. This quality analysis was instituted as a result of the findings between Perma, American Optical and Singer personnel September 3, 1964.

A random sampling of various dated shipments from American Optical was employed to obtain sensing devices for testing. These sensing devices, which totaled one hundred (100), were all checked and the findings approved by the parties attending.

The sample sensing devices were obtained from the shipments as follows:

Date of Shipment	Total Amt. of Shipment	Units From Each Shipment	Units From Each Carton In Shipment
7/7/64	3015	17	4
7/20/64	3015	17	4
7/24/64	3015	17	4
8/24/64	7284	34	4
9/1/64	3510	17	4

The component parts were checked visually and electrically according to the Quality Control check list of the Anti-Skid Unit. This check list was drawn up by Perma of which all three (3) parties received a copy at the September 8, 1964 meeting.

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Perma and American Optical disagreed on the method of checking micro switch settings. Perma preferred to check the micro switches with the solenoid assembled to the sensing device. American Optical preferred to check the micro switch with the solenoid disassembled from the sensing device.

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RESULTS OF TEST (9-8-64)

(Micro Switch Settings)

7-7-64 Shipment

No.	Solenoid Assembled	Solenoid Not Assembled	Remarks	Vacuum Bind
1	1 1/2	1 3/4		
2	1	2 1/2	Cam Retaining Spring (Part #122-C) not centralized over Cam Gear Button (Part #136-A)	
3	1 1/2	2 1/2		
4	1 1/2	2 1/2		
5	1 1/2	2 1/2		
6	2 1/2	3		
7	1 3/4	2 1/2	No clearance between Governor Shaft Seal (Part #119-E) and Delrin Governor Housing (Part #108-E) This unit used as cut away model.	
8	2 1/2	2 1/2		
9	1 1/2	2 1/2	Switch stays close with solenoid assembled	
10	1 1/4	2 1/4		
11	1 1/2	2 1/2		
12	1 1/2	2 1/2	Pin missing from Governor Housing (Part #110-G)	
13	1 1/2	2 1/2	Centrality of Gear Shaft (Part #137-C) and Gear Train Cover (Part #109-D)	
14	1 1/2	2 1/2	Bent Leaf Spring (part #122-C). Centrality of Gear Shaft (Part #137-3 and Gear Train Cover (Part #109-D)	
15	1 1/2	2 1/2	Switch stays closed with solenoid assembled	
16	2 1/4	3 1/2		Yes
17	2	3		Yes

7-20-64 Shipment

No.	Solenoid Assembled	Solenoid Not Assembled	Remarks	Vacuum Bind
1	1 1/2	2	No clearance between Governor Shaft Spool (Part #119-E) and Delrin Governor Housing (Part #108-E) Delrin Button (Part #136-A) cocked in Gear (Part #101-F)	
2	1 3/4	2 3/4		Yes
3	2	2 3/4		
4	2	3		
5	2 1/2	3 1/2		
6	2	3	No grease on Governor Shaft Spool (Part #119-#) or Ball Stud (Part #118-B)	Yes
7	2 1/4	3		
8	1 1/2	2 1/2	Cam Washer Shaft Pin (Part #RS-2) in backwards Friction Spring Retaining Pin missing. Thrust Washer (Part #RS-10) was not on Reverse Switch Cam (Part #RS-1) Cam Washer Shaft Pin (Part #RS-2) put in backwards.	
9	1 1/2	2 3/4	Centrality of Gear Shaft (Part #132-C) and Gear Train Cover (Part #109-D)	
10	1 3/4	2 1/4	No clearance between Governor Shaft Spool (Part #119-E) and Delrin Valve Housing (Part #108-E).	
11	5	5		
12	2	2 1/2		
13	1 1/2	2 1/2	Very little grease	Yes
14	2	2 3/4		
15	1 1/2	2 1/2		
16	1 3/4	3		
17	1 3/4	2 3/4	No clearance between Governor Shaft Spool (Part #119-E) and Delrin Governor	

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7-24-64 Shipment

No.	Solenoid Assembled	Solenoid Not Assembled	Remarks	Vacuum Bind
1	2	2 3/4		
2	1 1/2	2 1/2		
3	2 1/4	3 1/4		
4	1 1/2	2 1/2		
5	1 3/4	3		
6	5	5 1/2		
7	1 3/4	3 3/4	No clearance between Governor Shaft Spool (Part #119-E) and Delrin Valve Housings (Part #108-E)	
8	1/2	1 1/2	Micro switch will not open with solenoid assembled. Centrality of Gear Shaft (Part #137-C) and Gear Train Cover (Part #109-D)	
9	4 3/4	5 1/2		
10	2	3	No clearance between Governor Shaft Spool (Part #119-E) and Delrin Vlave Housing (Part #108-E)	
11	1 3/4	2 3/4	Centrality of Gear Shaft (Part #137-C) and Gear Train Cover (Part #109-D). Spring Thrust Washer (Part #RS-5) off. Reverse Switch Cam (Part #RS-1)	
12	1 1/4	2 1/2	Hesitates in going off	
13	2	2 3/4	Governor Spool (Part #119-E) tight.	
14	2	3		
15	2 1/4	3		
16	2	2 3/4	Switch Bracket (Part #133-C) cocked to one side of Cam Retaining Spring (Part #122-C). No clearance between Governor Shaft Spool (Part #119-E) and Delrin Valve Housing (Part #108-E)	
17	2	2 3/4		

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8-24-64 Shipment

No.	Solenoid Assembled	Solenoid Not Assembled	Remarks	Vacuum Bind
1	3	3 1/2		
2	3	3		
3	2	2 1/2		
4	2 1/2	3 1/2		
5	1 1/2	2	No grease on Reverse Cam Thrust Washer (Part #RS-5) in between Washer and Gear Train Cover (Part #109-D)	
6	2	2 1/2		
7	2	3 1/4		
8	2 1/4	3 1/2		
9	2 1/2			
10	1 1/2	2 1/2		
11	2 1/2	3	Reverse Cam Thrust Washer (Part #RS-10) was not aligned between pins	
12	2 1/2	3		
13	2 1/2	2 3/4		
14	2 1/2	2 1/2		
15	2	2 3/4	Centrality of Gear Shaft (Part #137-C) in Gear Train Cover (Part #109-D). No grease on Governor Spool (Part #119-E)	
16	2 1/2	3		
17	2	3	Bearing surface chopped up on Thrust Washer Bushing	
18	1 1/2	2 1/2	Centrality of Gear Shaft (Part #137-C) Gear Train Cover (Part #109-D)	
19	3	4	Reverse Switch Wire Contact (350-2) not soldered properly	
20	1 1/2	2 1/2		

8-24-64 Shipment Cont.

No.	Solenoid Assembled	Solenoid Not Assembled	Remarks	Vacuum Bind
21	2 1/4	3	Reverse Switch Contact Wire (350-2) not soldered properly	
22	1 1/2	2 1/2	Reverse Cam Thrust Washer Guide Pin busted.	
23	2	3		
24	1 1/4	4	Governor Shaft Thrust Washer too thin	
25	2	3		
26	2 1/2	3		
27	1 1/2	2 1/2		
28	2	2 3/4	No clearance between Governor Shaft (Part #119-E) and Delrin Valve Housing (Part #108-E)	
29	2	3		Yes
30	1 1/2	3		
31	2 1/2	3		
32	3	3		
33	2	2 1/2		
34	2	2 1/2		

[PX 38A, p 14]

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9-1-61 Shipment

No.	Solenoid Assembled	Solenoid Not Assembled	Remarks	Vacuum Bind
1	3	3		
2	3	3 1/4		
3	3	3		
4	2 1/4	3		
5	2 1/2	3		
6	2 1/2	3		
7	2	2 1/2		
8	2	3		
9	2	2 3/4		
10	3	3	Governor Shaft (Part #119-E) binds	Yes
11	1 1/4	2	Hesitates to go off with solenoid assembled	Yes
12	1 1/2	2 1/2		
13	2	2 1/2		
14	3	3		
15	2	3		
16	3	3 1/4		
17	2 1/2	3		

The findings of this quality control check are to be evaluated by Penna, American Optical and Singer and their conclusions presented at the meeting arranged for this purpose September 14, 1964.

H. Sharp

H. Sharp
Supervisory Engineer
Manufacturing Engineering
Contract Sales

J. Pecko

JC/ea

cc: H. Sharp
H. Snyder
J. Fecko
Those Attending

[PX 39, p 1]

PLAINTIFF'S EXHIBIT 39

Pltf's Ex 95-H for id 3/25/71 CS

Mr. J. Fecko, Manager
General Fabrication Division - Manufacturing Division

A. Romel, Manager
Department 195

September 17, 1964

Status Report - (Perma Anti-Skid Device)

Dear Sir:

The following is a list of problems encountered in the above unit and the steps being taken to solve them.

I. Hydraulic Pressure Switch

1. Leakage due to "O" Ring damage at assembly.

2. Switches set too high.

The problem of leakage has been traced to faulty manufacture and the use of second grade "O" Rings by the vendor. Recent samples received indicate that the vendor has corrected this defect. However, the samples furnished were found to be set too high when tested on our stand. In order to discount the vendor argument that our test equipment was faulty, we had two gauges calibrated and ran experiments to determine whether a pressure differential existed. This was done by mounting one of the calibrated gauges at the pressure switch and the other on the head board. No appreciable difference was noted during these tests and therefore we must conclude that our stand is reliable. Mr. J. Moore was instructed to notify the vendor and ask them to have their gauges calibrated and send in sample parts to be checked on our stand. To date no samples have been received and therefore this problem remains unsettled.

II. Sensing Unit

1. Rough finish on Governor Spool Shaft.

2. Mechanical bind.

3. Cam Gear fails to return.

[PX 39, p 1]

[PX 39, p 2]

Mr. J. Feeko, Manager
General Fabrication Division

September 17, 1954

At the last meeting held between Perma, American Optical and Singer, Mr. Larrousse stated that he would send a Quality Control representative to American Optical on Wednesday September 16, 1954 to perform a sample inspection of governor spool shafts. He indicated at that time that if his investigation showed that their inventory would meet the 32 R.M.S. finish required, he would assume the parts now at Singer would also meet this specification and therefore release the units for production after the other corrections were made. To date, no audit has been taken. I learned today that the plan now is to send someone to American Optical on Monday or Tuesday of next week. Aside from delaying this project, this postponement has given American Optical time to rework any defective shafts in their inventory.

Perma indicated that they would test several units that showed signs of mechanical bind to determine whether it would affect the operation of the unit. They are to let us know by September 17th if a separate test for this characteristic must be made.

Perma now feels that the cam track on the "Cam Gear" must be polished. They will be at Singer on September 17th to discuss this matter. This means all the units must be disassembled to perform this operation. Since American Optical has to remove the cover to set the switches it would be cheaper to have them perform this operation.

III. Adaptors

1. End play and runout of gear.

The vendor has been notified of these defects but insists that on some articles it is not spelled out on the print. They were to contact Perma and set up a meeting between the three groups to resolve the differences. To date, no meeting has been set.

IV. Gear Boxes

1. Premature wear.

A study of a gear box manufactured by American Optical showed that the parts were within print limits. The parts measured were assembled and run in for approximately 1,000 miles at which time the gear pin clearance. Upon examination

[PX 39, p 3]

Mr. J. Pecko, Manager
General Fabrication Division

September 17, 1954

it was found that the gear teeth were worn out and there were definite signs of over-heating. An analysis of the print tolerances indicated that several basic engineering principles necessary for good gear design were violated. The wearing characteristics on the gear teeth indicated that excessive dynamic load and improper mesh accelerated the failure. The presence of heat verifies excessive friction which caused overload of the teeth. Shearing of the gear pin is another indication of overload.

Several modifications are being made on the various components to determine whether the existing stock can be salvaged by replacing these parts. Six units will also be made to new figures and tested. Units made by "Swank" are being measured for comparison.

Representatives from Quaker City Gear have been invited to Singer on September 17th to review this problem. In the event that the existing gear boxes cannot be used at all, it may be wise to have a new box designed and built by these people. We would probably save time in taking this approach.

I am deeply concerned about the reliability of this device in general. My cause for alarm stems from the fact that the performance testing does not detect some defects which could cause a malfunction of the unit during operation on a car. I have discussed this matter with Mr. J. Wagner and it appears that we may be able to find a means of detecting these defects. However, a better understanding of the functional characteristics of the device must be attained before proceeding. I feel it would be advisable to get a detailed specification from Perma listing all the possible causes for failure so we may incorporate tests to detect deficiencies before sending units out of the plant. I strongly recommend that the legal aspects of responsibility be thoroughly investigated so that we may be fully covered for units sold before the confidence level is determined to be satisfactory.

Very truly yours,

A. Nemel, Manager
Manufacturing Engineering
Contract Sales

AR/ms

cc: Mr. C. W. Brocker

[PX 47, p 1]

PLAINTIFF'S EXHIBIT 47

Minutes of Meeting of September 24, 1964

Subject: Perma Anti-Skid Unit

In Attendance:

Messrs.: C. W. Brocker
J. Fecko
J. Larrousse
J. Moore
N. Hnojowy
R. Patten
H. Snyder, Jr.
R. Kloby
A. Kemel

A plan of action was requested by Mr. Brocker which would specify definite steps to be taken to solve the problems inherent in this unit. He also indicated that we should perform reliability tests on complete units to ascertain whether any new problems could be discovered which would cause malfunctions in the field.

A report was given by Mr. Larrousse on the results of his audit of governor spool shaft finishes at American Optical Company. He indicated that a large percentage of parts checked did not meet the 32 R.M.S. previously established by Perma. He also stated that he felt the units would function properly with the higher R.M.S. readings (approx. 40 to 63) found. Perma he said, is in agreement with this but wants to hold American Optical to the specified finish for bargaining purposes. Perma now wants the solenoid removed to lubricate the end of the governor spool shaft.

EXH. NO. 5A
BRAK. REPORTERS

As far as pressure switches are concerned, Mr. Larrousse reported that their audit at U. S. Dielectric showed the switches did not meet our quality requirements and therefore no shipping release was given.

The proposed changes on the cam gear were discussed and it was generally accepted that test information should be obtained before a change was made.

The gear box problem was discussed briefly and it was felt that the present course of action was satisfactory. This plan is outlined below.

As to the action to be taken in regards to the governor spool shaft finish and mechanical bind problems, there seems to be a considerable amount of confusion. My interpretation of what is to be done is listed below along with my personal comments.

Plan of Action:

I. Sensing Units

a. Governor Spool Shaft Finish

Mr. Larrousse indicated that he would perform another sampling to check this characteristic as well as the mechanical bind. The data is to be presented at a meeting to determine a course of action.

I personally feel that the sample of 100 units taken at our plant and the audit performed at American Optical has provided enough information to determine whether the lots meet specifications or not. As far as this problem is concerned I feel a definite position must be obtained from

Perma as to whether the parts with the higher R. M. S. readings can be used.

As far as "mechanical bind" is concerned I was under the impression that it was agreed that this was not a major problem. We were to spot check this characteristic to detect possible finish defects in component parts.

II. Pressure Switches

a. A cost for disassembly, repair, cleaning, reassembly and setting at Singer is to be made. Singer will perform these operations and charge back the cost to U.S. Dielectric if they do not correct these units.

III. Cam Gear Changes

a. Tests will be made on:

1. Cam gears polished only.
2. Cam gears with cut down over-ride only.
3. Cam gears with cut down over-ride and polished.
4. Cam gears with increased helix angle.

A decision to change the mold will be made based on the above findings.

IV. Gear Boxes

a. Sample of a new design is due on September 25, 1964 from R. M. Stewart Inc.

b. Quaker City Gear Works is to submit six altered boxes on September 30, 1964 in an effort to salvage existing stock. They

will also propose a new design if they feel it is needed.

c. Our efforts to modify the present stock will continue. Six sample boxes are being altered for testing.

d. A variable speed motor test set-up is being built to life test these units.

V. Adaptors

a. Stock is being sampled by Quality Control; defective lots will be returned to the vendor. Perma has agreed to change some of the tolerances and will be in to present new drawings the week of September 28, 1964.

VI. Product Reliability Testing

Several complete units will be tested with the facilities now available. These we feel are not adequate and therefore life testing equipment and procedures must be set up in the near future. I feel engineering and quality control must jointly develop the techniques to be used. I have requested information from Perma which will point out the critical areas of the unit. This information is necessary to insure adequate testing is provided. I propose that once the equipment and procedures are set up and functioning properly that the product reliability testing become the sole responsibility of quality control.

[PX 47, p 5]

A. Romel, Manager
Manufacturing Engineering
Contract Sales

AR/ms

cc: Those attending
Mr. H. Sharp

[PX 49, p 1]

Ref # 11-9-73 DEF. EX. 107

PLAINTIFF'S EXHIBIT 49

SINGER

321 FIRST STREET • ELIZABETH, N. J.

MEMORANDUM

TO: Mr. C. W. Brocker, Assistant General Manager
Manufacturing Division

FROM: A. Romel, Manager - Department 195

DATE: September [illegible] 1964

REFER TO:

SUBJECT: Status R[illegible] "Perma"

I. Sensing Units

a. Governor Spool Shaft Finish

11/9/73

On September 25, 1964 Mr. J. Perrino informed Messrs. R. Kloby, J. Moore and the writer that "Perma" could not accept the units with the higher R.M.S. readings until results from their current tests are available. Mr. Moore informed American Optical of the following position to be taken by Singer.

American Optical must remove the solenoid and inspect for finish. They are to ship only those units that meet the 12 R.M.S. finish. They are to lubricate this surface at this time for which they will be paid. The units that do not meet the finish requirements are to be held pending the test results at "Perma". If the tests are positive a release for the remaining units will be given. Singer will sample lots received and reject on a 2-1/2% A.Q.L. basis.

Mr. Moore reported that American Optical's reaction was non-committal and that they must check with their legal department before giving a definite answer.

[PX 49, p 2]

It should be noted that Mr. F. Perrino will authorize Singer to accept the rejected parts if it is felt that we must back down from this strong position. Since there are other unresolved problems preventing production it is not unreasonable to utilize the interim period to negotiate this item.

b. Cam Gear

1. Polished only	-	Proved 50% reliable on our test stand.
2. Cut down override only	-	(2) Gears tested at 40 m.p.h., stop failed to return immediately. At 60 m.p.h., both functioned properly.
3. Cut down override & polished to 20-22 R.M.S.	-	No failures in (5) gears tested at 40 m.p.h. stop test.
4. Increased helix angle	-	No samples available for testing.

The above tests indicate that cam gears in item #3 are reliable and therefore the mold will be changed. The new print now specifies a 32 R.M.S. finish polished to 20 R.M.S. An ultimate design with a larger helix angle will be pursued to eliminate the variables that are inherent in the polishing operation. Mr. Perrino agreed that this would be done at some future date but could not be done immediately in that this change affects other parts. Calculations for new switch settings and other changes will be looked into.

[PX 49, p 3]

- 3 -

Mr. Moore received a delivery date of two (2) weeks for a sample lot of 2,000 parts with production runs in six (6) weeks. He will try to better the production commitment or increase the sample lot size.

II. Pressure Switches

a. Cost figures for reworking were given to Mr. Moore.

	LOT SIZE 2.5M	5M	10M
Full absorption	$66.86/C	$65.27/C	$64.00/C

Mr. Patten instructed Mr. Moore to use $54.00/C. The tool cost was estimated to be $160.00.

b. Mr. Snyder is waiting for authority to rework a quantity of these parts.

c. It now appears that we should take steps to tool-up for this item at Singer in that the present vendor is not reliable. Since there may be tooling available from the suppliers, engineering will not take action to provide new tooling unless so authorized by purchasing.

III. Gear Boxes

a. The sample received from R. W. Stewart was not made to the specified speed ratio and therefore cannot be used. They were notified and are investigating to determine whether this type of design could be made to meet the specification desired.

b. Quaker City Gear Works still feels they will have samples by September 30, 1964.

c. One (Singer) altered gear box is ready for life test. Five other boxes should be ready by October 2, 1964.

d. The test set-up is being held up by Mr. Wagner in that he has not located the variable speed motor as yet. He was instructed to buy one if it is not located by Monday, September 28, 1964.

IV. Adaptors

No report received from Quality Control by this office to date.

V. Product Reliability Testing

a. Components are being checked before assembly of the complete units. A quantity of (12) units is to be assembled.

b. No contact has been made with Quality Control to date due to changes in their organization. It is my understanding that Mr. Plungis will now be responsible for the quality control function of this division. He will be contacted as soon as this becomes official.

VI. Perma-vac Unit

A new design for the check valve spring, Part No. 17-342000 was submitted by Perma. Mr. Moore was given a drawing and will advise as to when new springs

would be available. Mr. Snyder was informed that the old springs are to be replaced and shipped to "Perma". Perma to be charged for this change.

A. Romel

A. Romel, Manager
Manufacturing Engineering
Contract Sales

AR/ms

cc: Messrs. J. Fecko
H. Snyder, Jr.
J. Moore
R. Kloby
R. Patten
H. Sharp

NO MATERIAL ON PAGE E-430

PLAINTIFF'S EXHIBIT 53

RECEIVED
DEF. EX. 120.
DEC 7 1964
MANAGER'S OFFICE

Mr. J. Pecko - Manager - General Fabrication Division

A. Romel

December 4, 1964 AR:ms

"Ferm" Sensing Units

Dear Sir:

Inspection of the sensing unit removed from the Singer station wagon revealed that grease from the gear box and/or transmission is being pumped into the unit through the drive cable. We have complained about this to Ferm representatives on numerous occasions but they have not been able to come up with a suitable design change. This condition first came to our attention when we started testing gear boxes. The amount of grease and graphite noticed during these tests were comparatively small. The unit taken off the station wagon was packed with so much grease that it no longer cycled. I have discussed this problem with Mr. Brecker and he indicated that steps would be taken to stimulate some interest in this problem by Ferm.

Our engineering staff has been working on this problem but have been restricted by the present design. We will continue our efforts to resolve this problem.

The unit taken off the test car is available for your inspection.

Very truly yours,

A. Romel

A. Romel, Manager
Manufacturing Engineering
Contract Sales

cc: Messrs. C. W. Brecker
D. A. Adams
R. Kloby
H. Hutton
J. Moore
H. Sharp

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PLAINTIFF'S EXHIBIT 54

perma

Memo: Perma Anti-Skid Unit

Subject: Discussion between Perma and Singer Personnel.

Date: July 17, 1964

In Attendance:

H. Snyder
R. Kloby
E. Kinal
S. Margerum
J. Gomes
J. F. Moore, Jr.
R. Minelli
F. Di Benedetto

1. Check Valve Body (17-342078) (Perma No. PV-9)

   a. The majority of the check valve bodies are oversize and will not fit into the connector block.

   b. Agreed to ream out connector blocks to .599 for all 5000 check valve bodies on hand.

   c. Singer will absorb cost of this added operation.

   d. Future deliveries of check valve bodies will be returned to vendor if they are oversize.

2. Check valve assembly (17-342004) (Perma No. PRA-905)

   a. There must be a minimum of 1/16" movement of the check valve seat. This must be shown on the assembly drawing.

   b. When the Valve piston (17-342008) (Perma No. PV-2B) is assembled to the Check Valve such that the nose of the valve piston is just touching the seat of the check valve, there must be a clearance of 1/16" between the faces of the check valve body and valve piston.

D 125

3. Check Valve Seat (17-342079) (Perma No. PV-8)

   a. Agreed to change surface finish from 16 R.M.S. to a 25 R.M.S.

   b. All check valve assemblies on hand will be accepted provided that they pass the leak test performed at 200 p.s.i.

   c. Since it requires disassembling the check valves to inspect the surface finish on the seat it was agreed that Mr. Moore would negotiate with the vendor to have the assembly performed at Elizabeth.

4. Pressure Switch (17-342025) (Perma No. SPR-1100A)

   a. The specifications were altered as follows:

      1. With the pressure decreasing the switch contacts should open between 85 and 100 p.s.i. instead of 85 to 95 p.s.i.

      2. With the pressure increasing the switch contacts must be closed before the pressure reaches 140 p.s.i.

   b. The inventory of pressure switches received from Worcester Stamp Metal were not properly set. It was agreed that we will estimate a cost of resetting the pressure switches. This information will be given to Mr. Kloby who will deal with American Optical to negotiate some agreement whereby we will reset each switch.

5. Piston Body (17-342000) (Perma No. PV-1A)

   a. Agreed that all piston bodies that do not meet with the 16 R.M.S. finish on the I.D. will be rejected and returned to vendor. Mr. Moore will contact vendor and advise them of reason for rejection.

6. Gear Box (17-342054) (Perma No. VPR-800)

   a. Gear boxes received from Worcester Stamp Metal inventory cannot be used. They are to be set aside until Perma determines disposition.

   b. Mr. Minelli agreed to send Engineering the layouts he has of the gear box.

7. Anti-Skid Device (17-342034) (Perma No. SPR-1000)

   a. Perma has written a letter to Mr. Patten calling for a design change. This change adds a concave washer (Singer Part No. 97-20469, tumbled and case hardened) to the reverse switch mechanism to eliminate the problem of the bee-hive spring wearing out.

   b. Mr. Snyder agreed to send 15,000 of 97-20469 to Departments 33 and 20. These will be used to alter the Anti-Skid Devices that are on hand and will also be sent to American Optical for altering their stock, as well as using them on the current production.

8. Perma-vac press metal components made by Worcester Stamp Metal.

   a. It was agreed that Perma will initiate action to pull all of the press metal tools out of Worcester Stamp Metal for manufacturing the following components:

      1. Cover (17-342005) (Perma No. PV-6C).

      2. Piston Plate (.095 thick) (17-342011) (Perma No. PR-323).

      3. Piston plate (.062 thick) (17-342014) (Perma No. PR-326).

      4. Piston plate (.025 thick) (17-342015) (Perma No. PR-331).

      5. Container (17-342066) (Perma No. PV-6)

9. Perma-vac unit complete (17-342033) (Perma No. PRA-510)

   a. Agreed to place plastic plugs in the hydraulic fitting to prevent dirt from entering the hydraulic system.

J. Gomes
Assistant Chief Engineer

JG/ma

cc: All participants,
Messrs. N. Hnojowy, R. Patten, A. Romel

PLAINTIFF'S EXHIBIT 57

M-44

WILLIAM E. HILL & COMPANY, INC.
MANAGEMENT CONSULTANTS
NEW YORK LONDON BRUSSELS

640 FIFTH AVENUE
NEW YORK 19
JUDSON 2-5959
CABLE: HILLMANCO NEW YORK

December 14, 1964

Mr. Robert A. Kloby
Director of Forward Planning
The Singer Company
Elizabeth Plant
321 First Street
Elizabeth, New Jersey

Dear Mr. Kloby:

In accordance with your assignment, a preliminary survey of short-term prospects for the Perma anti-skid control has been completed. The following report summarizes the findings and conclusions of this survey, which were reviewed with you November 11 in Elizabeth.

Sincerely yours,

William E. Hill & Company

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11/20/70

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14/14/64

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The Singer Company

PRELIMINARY MARKET SURVEY

PERMA ANTI-SKID CONTROL

The objective of this project has been to assist management in determining the short-term market prospects for the Perma anti-skid control. Although determination of sales potential was the principal objective, the issue of product performance arose during the course of the project as an important consideration, and the subject has been covered in the report.

Following orientation meetings with Singer and Perma management, the conduct of the survey included meetings with key industry sources such as automotive brake and safety engineers, auto manufacturer marketing personnel, fleet operators and safety engineers, a fleet operation consultant, specialty automotive part distributors, new-car dealers, and several Perma distributors.

The principal findings and conclusions resulting from this preliminary study are summarized below.

1. Automotive engineers have recognized for many years that it is possible and highly desirable to improve the braking operation by adding to the vehicle braking system a mechanism that

would sense an impending locked-wheel condition and prevent the brakes from coming to a completely locked condition. A system which could do this would allow the car to be stopped somewhat faster and give greater steering control in a panic stop situation. The Perma control is one of several systems that automotive engineers have evaluated in recent years in their search for a system which will give the desired improvement in braking performance at a reasonable price.

2. The Perma anti-skid control falls short of meeting requirements of automotive engineers, and does not provide the improvement possible in theory. The consensus of the many engineering tests ve been run on the unit indicate that the Perma contro , as compared to a panic or locked-wheel stop, gives improved steering control, but requires a greater distance to come to a complete stop. The automotive brake and safety engineers who have reviewed its performance do not agree on the value of the Perma anti-skid control, and are about evenly divided on the issue. One group believes that it is of value in that most drivers cannot handle a skid situation adequately and that the added control of the car obtained is well worth the relatively small loss in stopping distance. The other group believes that no amount of increase in control justifies the increased stopping distance.

3. The user aftermarket represents the only apparent short-term opportunity for significant sales volume for the Perma anti-skid control. Prospects are indicated for a sales volume on the order of 100 thousand units during the next two years, depending on sales effort.

   a) The product, with its strong emotional appeal to safety, and with demonstrable characteristics, lends itself to a consumer-directed merchandizing effort. Typical of products of this type, sales volume will depend largely on the amount of promotional effort expended.

   b) The original-equipment market represents virtually no opportunity at this time for sales of the Perma control. In addition to the mixed opinion on its worth, this is because of the low value the automotive manufacturer places on safety equipment, considering the poor record of consumer acceptance for high-cost equipment such as the Perma control.

c) The prospects for significant sales to fleet markets are expected to be quite limited due to the highly cost-conscious nature of fleet operations. Most government fleets are subject to stringent budgetary controls with very little allowance for extras. Large professionally run private fleets subject their expenditures to careful cost analysis, and the anti-skid control can seldom be justified on a dollars and cents basis, considering the substantial investment cost in relation to reduction of accident cost.

4. To meet the sales volume which appears possible during the next two years, a strong promotional program aimed at the user aftermarket would be required.

a) An intensive promotional program would be needed, because the Perma anti-skid control is virtually unknown and would have to be introduced to the end-user. In addition sales would represent discretionary purchases (versus replacement parts sales) and the item is costly in relation to most other auto accessory purchases.

b) A program to sell the item successfully would likely depend heavily on "high-pressure" promotional methods which include a high sales and advertising cost per unit, sales practices capitalizing on a strong emotional appeal, and perhaps emphasizing strong "personal" selling.

c) Any one of a number of channels of distribution could be effective for this product, including new-car dealers, gas stations, specialty auto accessory stores and a direct selling organization. However, to meet the desired sales volume through any one of these channels, it appears that a supporting program of advertising, demonstrations and other promotional measures would be required.

5. If the Company becomes involved in a sales program directed to the user aftermarket, management will want to recognize the possible problems which might arise, and which have been encountered by other companies in similar situations. In particular, the following points might be considered.

a) Use of the Singer name both at dealer and user levels should be carefully controlled. This is essential to prevent misuse of the name in relation to extravagant product claims and also to prevent association in the user's mind with possible undesirable sales and promotional activities.

b) Management, of course, will want to determine the legal liability which could arise from accidents where it could be claimed that the anti-skid control did not function as represented or where it may have been installed incorrectly or become inoperative.

c) In light of Singer's dependence on an independent sales organization for marketing of the Perma control, management will want assurance that the program will not resort to the use of door-to-door sales organizations which would employ high-pressure salesmen. This in particular could result in problems possibly harmful to the Singer name and reputation.

d) The question of reliable repair service and replacement parts availability is an important one, and assurance of reliable resources in this respect will need to be considered.

The following section of this report summarizes in detail the findings of the survey.

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The Singer Company

PRELIMINARY MARKET SURVEY

PERMA ANTI-SKID CONTROL

A. Product Description and Engineering Evaluation

1. For many years automotive engineers have recognized that it is desirable as well as possible to reduce stopping distance of a car in a panic stop by holding brakes steady just before a full locked-wheel condition and at the same time reduce the tendency for the car to spin. In recent years many of the automotive companies have evaluated systems which are able to make this improvement in braking performance at a justifiable price. The Perma anti-skid control is one of the devices which have been designed to make this improvement in braking performance. This control operates by quickly sensing a locked-wheel condition and momentarily reducing the brake line pressure to allow the wheel to rotate. The system cycles through this lock-roll-lock-roll condition about four times a second until the car comes to a halt.

a) In theory, by holding the wheels at a 20 percent skid, or by allowing the motion of the car to come 20 percent from skidding and 80 percent from wheel rotation, the car can be brought to a stop faster and kept under better control than in the locked-wheel condition.

b) The lack of stability, or the tendency for the car to swing around, is particularly serious in the case of Bendix-type brakes which are self-actuating and quite sensitive to the condition of brake lining and adjustment for balanced braking. Disk brakes, which several U.S. manufacturers are offering this year as optional equipment, operate on a somewhat different principle and do not have as great a problem in braking stability as self-actuating types.

c) In addition to the evaluation made of the Perma anti-skid

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control, automotive manufacturers have evaluated the Lockheed and Bendix systems. Also, there is a comparable system developed in Europe which was offered on the Jaguar auto as optional equipment. The Lockheed unit operates on the basis of a small flywheel backlash operating through a pneumatic system to reduce brake pressure; the French system is a mechanical system working on each wheel.

2. A number of engineering tests have been run on the Perma anti-skid control comparing its stopping performance in terms of stopping distance and stability with the operation of the car under locked-wheel conditions. The consensus of the tests is that the Perma control allows a car to be stopped under panic stop conditions with better control, but at the cost of greater stopping distance when compared to locked-wheel conditions. While this conclusion is definitely substantiated by most of the results, there is some variation between the test results. The results of tests run to date have been summarized in Exhibit 1.

   a) Exceptions to the above conclusions are as follows:

      (1) The test results of the Motor Vehicle Research group, which were sponsored by Perma, showed that under all road conditions use of the Perma anti-skid control allowed the car to stop significantly faster than under locked-wheel conditions.

      (2) In the Cadillac Division test, under glare ice road conditions the Perma control permitted the car to stop in slightly less distance than the locked-wheel situation.

      (3) The Ford tests showed that the car equipped with the Perma control took longer to stop and gave no significant improvement in stability.

   b) The extent of the testing performed in evaluating the control was reviewed with the auto manufacturers' engineers conducting the tests in most cases, and the amount of time and expense involved was considerable and results are considered to be sound.

3. There are two points of view among the engineers evaluating the Perma anti-skid control as to the value of a system which brings a car to a stop with better control but which requires greater

distance to stop; the engineers are almost evenly divided for and against use of such a system.

a) The General Motors Research Center, The Ford Advanced Design Group and Chrysler Brake Laboratory are against use of the control. They believe that nothing should be done to increase the stopping distance required under panic stop conditions, because the last ten feet may be the critical ones in a collision.

b) American Motors Safety Engineering and Cadillac Division Brake Design Group are in favor of the system. They believe that the average driver will not remember to pump his brakes under panic conditions and hence is much better off with the anti-skid control. He will have enough control to avoid most road obstructions and stay in a controlled path even though it takes him slightly longer to stop.

B. Possible Markets for Anti-Skid Systems

There are three distinct markets representing significant potential for products such as the Perma anti-skid control.

1. Original equipment manufacturers, represented by the major automotive companies, General Motors, Ford, Chrysler, and American Motors.

2. Fleets, including privately-owned fleets and government-operated vehicles.

3. The automotive user aftermarket, represented by the sales of parts and accessories after the car is purchased, and including accessories installed by the new-car dealer.

C. Market Characteristics and Requirements

Following is a summary of the key market requirements relating to automotive equipment such as the anti-skid system.

1. The automotive manufacturer has two essential requirements,

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dependability and value. These requirements are reflected in the typical evaluation of a possible new device; first, assurance that the device is sound from an engineering point of view, and secondly, consideration as to profit, which is a function of user acceptance (in terms of the added price he is willing to pay) and the added cost to the manufacturer. Safety features have typically received low ratings in relation to these criteria.

a) The automobile companies are slow to accept new accessories because of the desire to thoroughly prove out any new device for reliability. For example, the power-braking system on the Ford, similar in complexity to the Perma anti-skid control, took over two years to develop even after the basic design was well established. All new accessory designs have hundreds of thousands of miles of testing before final acceptance. In this respect the statement was made by several of the auto companies that from a reliability point of view they believe the Perma anti-skid control is below their standards.

b) An example of the rather low rating that auto companies put on safety items is the case of dual brakes. This item has been put only on the Cadillac and Rambler; it permits partial brake operation even though a hydraulic line is broken. Although the cost is low, about $1.50 in manufacturing costs, other manufacturers have not added the device, primarily because they do not believe the consumer will recognize that amount of additional value.

c) While it is difficult to find a product which is exactly parallel to the anti-skid control, the record of seat belts probably comes closest. These cost in the $25 to $35 range during their first year. Through the 1950's American Motors was the only manufacturer to offer belts as an accessory, and although belts had unqualified support of safety organizations the company realized sales on under 1 percent of cars until the legislation of the 1960's.

2. The requirements of fleet markets in total center around cost consideration, particularly in the case of government and major fleet operations. The nature of these cost requirements are different, however, between the two major market segments.

a) The key characteristic of government fleet purchasing is the overwhelming concern with price. Virtually all purchasing of motor vehicles and accessories is subject to public bidding

and review. Hence, except where special equipment is required to accomplish a particular job, the government vehicle is strictly transportation and the cheapest equipment available is purchased.

(1) The Federal Government fleet has the most stringent financial limitation placed on it. All of their vehicles, with accessories, are purchased for under $1,500. This leaves no allowance for anything but the most necessary accessories. This ceiling has been set by law and there is no indication that it will be raised in the near future.

(2) The GSA, which purchases and operates the largest share of government vehicles, has shown interest in many new safety devices and has been discussing these devices with automotive companies. However, the agenda of their last meeting in Detroit this November did not include the anti-skid control, other devices being considered of higher priority.

(3) Smaller government fleets, such as local police vehicle fleets, have indicated interest in the anti-skid control. Where vehicle purchases are not subject to close public scrutiny and key persons influencing the sale may have a freer hand in choosing the equipment, the cost factor may be secondary. A number of these smaller fleets have shown initial interest in the system and the Perma control has been installed in a few trial cars.

b) The principal concern of the commercial fleet is cost of operations. These cost considerations fall into two areas: first, the cost of depreciation (the relationship between purchase price and eventual resale on the used-car market) as it affects the car and the accessory equipment, and second, the cost of operation including fuel, maintenance, insurance repairs, etc. In the large professionally managed fleets cost is the primary concern, although allowance is often given to match fleet equipment with the personally owned vehicle the driver is accustomed to.

(1) The large fleet operator looks at his costs as divided quite equally into three parts -- depreciation, direct operating costs and all other, including the costs of accidents. Of the total cost of operation the direct cost of accident (the sum of insurance and non-insured accident costs)

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represents a minor 9 percent in the average business fleet. With this relatively low base, very little cost for accessory equipment can be justified in an attempt to reduce it.

(2) The costs of accidents are normally 85 percent covered by insurance, and unlike private auto insurance, the premium rate is negotiated annually with the underwriter. The rate varies from fleet to fleet, but is nearly always based on the accident experience of the previous year. Hence the large fleet operator will only consider installing a safety device when it can be shown that it will reduce the accident cost in relation to its cost.

(3) The fleet owner often will purchase accessories which have value from an employee relations point of view. The rule of thumb on this type of purchase usually is to give the driver of a company car equipment he has come to accept in his own car. Hence, after power brakes rose to a rate in new car installation of around 50 percent the large fleets started to purchase cars with power brakes. The same situation has occurred with the automatic transmission and the 8-cylinder engine. On this basis the fleet operator would not purchase a new accessory until after it had received fairly wide consumer acceptance.

(4) The cost of depreciation over the time that the fleet holds a car, usually two to four years, is a factor in choice of accessories. The fleet operator recognizes that the accessories depreciate in market value much faster than the car itself. In fact, only five accessories are recognized as having resale value; these are automatic transmission, radio, power brakes and steering and air conditioning. All others, including Perma anti-skid control, are nearly valueless on the used-car market. This fast rate of market depreciation would be a further barrier to sales in this market.

(5) In addition to the large professionally managed fleets there are, of course, small fleets that are not subject to close cost control. These fleets are often run by one man with a relatively free hand over purchases, and who may recognize unusual operating factors such as poor road conditions. Many of these operators may be in a

position to purchase a device such as the anti-skid control simply because the key man is convinced of its value.

3. There are a number of important requirements which would have to be met to successfully sell the Perma control in the user aftermarket.

a) Promotion to the user is the most important market requirement. The anti-skid control, in spite of the considerable amount of notice received in trade press, is virtually unknown to the car-owning public, and its story must be explained. In addition, the control is an item that a car owner does not have to have, and it therefore must be promoted more intensely than service or replacement parts which are necessary to operate his car. Promotional techniques which might be used include television "spot" announcements, group demonstrations, movie shorts, newspaper and magazine advertising and personal calls.

b) Demonstrated reliability recognized by the motoring public would be essential to marketing of this product, both because it is new and because it becomes a part of the braking system of the car. Reliability could be demonstrated in a number of ways, including association of the control with established and universally respected names such as Singer. In addition, the record of fleets or other groups that have had good field experience with the device would be helpful.

c) Installation service for the unit must be competent. Although the unit does not require much time to install, the person doing the job must be skilled. Because the unit goes into the braking system, all installation work must be sound.

d) Credit would be needed for this item because the relatively high cost of the Perma control cannot be met by the typical aftermarket purchaser in a cash payment.

D. Distribution to the User Aftermarket

Three established distribution systems represent possible channels for sales of the Perma anti-skid control, and meet to varying

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degrees the requirements for serving the user aftermarket. A fourth means of distribution -- direct-to-user sales -- would represent a fourth possibility. Effective use of any of these distribution systems would require strong promotional support.

1. New-car dealers presently sell independently manufactured accessory items as part of the new-car sale or when the new car is serviced. The dealer operates in a highly competitive environment in his sales of new cars. For instance, the average Ford dealer makes only $27 on each new-car sale, with General Motors dealers only slightly better. Accessories, on the other hand, have a much higher margin and present the dealer and his salesmen with a better profit opportunity than the car itself. New-car dealers would be interested in handling the anti-skid control as a relatively high-profit accessory item, and Perma's New York distributor appears to have had some limited initial success in placing the item with dealers.

2. The gas station, particularly selling through the TBA (tires, batteries and accessories) divisions of national oil companies, would present a means of distribution. The typical gas station is faced with a highly competitive situation in sales of gasoline and the TBA program is an effort to broaden the station's sales into higher-margin items. This channel is of particular interest because of the favorable reaction to the unit developed with Phillips Petroleum, which has been most interested in promoting the item. The gas station is able to offer credit through their credit cards. Although many stations would have the necessary installation competence, small stations do not have the skill to install the unit. In addition, the gas station is not a promotional outlet and would have to be backed with an independent promotional program.

3. The independent automotive specialty chains could possibly present a good outlet for the Perma control. These selling and installing organizations include U.S. Royal, Firestone, B.F. Goodrich, Vanderbilt, Midas Muffler and others. These outlets base their business on strong promotion and advertising and could handle the anti-skid control. The issue with this

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type of distribution is that they are very selective as to products handled. Their objective is to handle products which will make most efficient use of salesmen's time, and hence prefer established products for which user acceptance is well established and which can be sold with the minimum of sales time and effort. Also, in line with the nature of these operations, there is a requirement for rapid turnover of stock; one major chain sets a minimum of four times a year. This limits the number of different products these stores will carry.

4. Direct user sales has been shown to be a possible method of selling anti-skid control. Several of the Perma distributors have had limited experience in selling through a number of direct-sale operators. Techniques have included demonstrations, following leads from advertising or from promotional talks, and neighborhood gatherings. This type of selling gives an opportunity for very heavy promotional effort. Arrangements would have to be made independently by the salesman for installation and credit. Large businesses in other products have been developed on this distribution basis and it can be a highly effective method of selling certain specialized products.

E. Short-Term Market Prospects

Based upon the results of this preliminary survey, it appears unlikely that the stated sales objective for the Perma control will be attained. With the prospect for sales to the automotive manufacturers virtually nil in the foreseeable future, and limited prospects in the fleet market, significant sales volume during the next two years will have to be realized from the user aftermarket. An aggressive, promotionally-oriented marketing program could probably develop sales of approximately 100 thousand units within the next two-year period. Estimates of sales

to the principal markets are summarized in Exhibit 2.

1. Based on evaluation by major automotive manufacturers, the Perma anti-skid control does not meet established requirements, and cannot be expected to find a place in the OEM market under present conditions. In the longer-range future, however, it is possible that anti-skid controls of some type may be included in the braking system of automobiles as original equipment.

2. Despite the large number of vehicles operated in civilian and government fleets, the sales potential for the Perma control appears quite limited. It is estimated that approximately 500 units could be sold the first year, and from 2,000 to 4,000 the second year. It is expected that these sales would be to small civilian fleets. The strict cost considerations under which professionally run fleets operate cannot justify a high-priced item like the Perma control, and impose a severe limitation to sales prospects in this market.

3. Depending on the effectiveness and intensity of sales effort, it is possible that on the order of 25,000 units could be sold in the user aftermarket during the first year, and perhaps two-to-three times this volume the second year, or from 50,000 to 75,000 units. It is most likely that these sales will be through car dealers and direct-sales organizations. TBA and specialty automotive products companies contacted as part of this survey expressed strong reluctance to taking on the Perma control -- mainly because their operations are not geared to promotion and introduction of new products.

4. Although it is beyond the scope of this preliminary survey to evaluate in depth the various distribution methods open to the Perma control, or to recommend a marketing program, it is apparent that the anti-skid control has certain characteristics which could support a major direct-selling operation.

   a) Involving, as it does, family safety, the anti-skid control is subject to promotion on a strong emotional basis. Coupled with this is the fact that it is a product which is demonstrable in familiar and understandable terms to the large car-owning population. Selling at a price well over

$100, there is a good margin for significant sales commissions.

b) In recent years a number of consumer products having some of the above characteristics have been very successfully sold on a door-to-door basis and sales volumes so developed have been impressive at the peak of such programs. Sales volume is limited virtually only by the number of salesmen employed. Products such as water softeners, home fire and burglar alarms, storm windows, siding, and vacuum cleaners are examples of products sold by this type of sales operation.

c) Many of these selling organizations have engaged in the doubtful practices to which direct sales activities are subject, and have experienced serious financial and legal programs. In this regard estimates of possible sales of the Perma control have assumed good control of sales and promotional activities and careful selection of sales representatives. Marketing by means of the typical door-to-door operators on an unrestricted basis would represent a very different set of conditions and could present much greater sales prospects.

Exhibit 1

RESULTS OF ENGINEERING TESTS OF THE PERMA ANTI-SKID CONTROL

Agency Sponsoring Testing	Location	Approximate Results of Stopping Distance Tests		Stability of Car During Locked-Wheel Stop	Reliability of Unit	Overall Estimate of the Worth of the Unit
		Dry Pavement	Wet & Slippery			
Perma Research & Development	Motor Vehicle Research (Maine & New Hampshire)	15% less stopping distance with Perma	5% to 20% less distance with Perma	Considerable increase in stability	Laboratory tests of certain parts of the unit show reliability	Very valuable
Cadillac Motor Car Co. (div. of General Motors Corp.)	Delco Moraine Div. of G.M. (Dayton, Ohio	5% to 10% more distance required with Perma	Slightly less stopping distance with Perma	Increased stability	Not tested	Has some value in control of car even though longer stopping distance
General Motors Technical Center	G.M. Proving Ground (Detroit)	9% to 18% more distance required	4% to 27% more distance required	Increased stability	Not tested	Gain in stability does not justify longer stopping distance
Chrysler	Chrysler Test Facilities (Detroit)	5% to 15% more required	5% to 15% more required	Increased stability	Not tested	Gain in stability does not justify longer stopping distance
Ford	Ford Test Facilities (Detroit	1% to 10% more required	5% to 10% more required	No significant increased stability	Not tested	Not of value

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WILLIAM E. HILL & COMPANY, Inc.

Exhibit 1 (Cont.)

RESULTS OF ENGINEERING TESTS OF THE PERMA ANTI-SKID CONTROL

Agency Sponsoring Testing	Location	Approximate Results of Stopping Distance Tests		Stability of Car During Locked-Wheel Stop	Reliability of Unit	Overall Estimate of the Worth of the Unit
		Dry Pavement	Wet & Slippery			
American Motors	No formal tests made Unit on test car in car pool	-	-	Increased stability recognized	Not tested	Has some value in control of car even though longer stopping distance
Phillips Petroleum	Phillips Test Track (Bartlesville, Oklahoma)	-	5% to 20% more required	Increased stability	Not tested	Unit has value in control of car; would also like to see shorter stopping distance

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Williams Printing Company, Inc.

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Exhibit 2

Perma Anti-Skid Control

PRELIMINARY ESTIMATE OF MARKET POTENTIAL

SUMMARY BY PRINCIPAL MARKETS

Market	Key Factor in Selling Market	Rate of Penetration	Potential Unit Volume 1st Year	Potential Unit Volume 2nd Year
Original Equipment Manufacturers	(1) Engineering acceptance	None expected	-	-
	(2) Cost versus value on market	None expected	-	-
Fleet				
Civilian	Reduction in operating costs	Slow	500	3,000
Government	Low initial cost of article and equipment	Very slow	150	1,000
User After-Market	Strength of promotional effort	Potentially rapid, depending on promotional program	25,000	70,000